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SIX-FOOT-DIAMETER MULTICYCLE METALLIC DIAPHRAGMS FOR SUBCRITICAL CRYOGENIC FLUID STORAGE AND EXPULSION

David Gleich
Arde, Inc.

TECHNICAL REPORT AFAPL-TR-70-95
February 1971

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Air Force Aero-Propulsion Laboratory
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AFAPL-TR-70-95

SIX FOOT DIAMETER MULTICYCLE METALLIC DIAPHRAGMS FOR SUBCRITICAL CRYOGENIC FLUID STORAGE AND EXPULSION

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FOREWORD

This report was prepared by Arde, Inc., Mahwah, New Jersey under USAF Contract No. AF33(615)-2827 which was in effect from July 1965 to December 1970. The contract was initiated under Budget Program Sequence Number (BPSN): 5(63 3145 624 05214), "Subcritical Cryogenic Expulsion System, Metallic Expulsion Diaphragms". Principal contractor investigator was David Gleich. The work was administered under the direction of the Air Force Aero-Propulsion Laboratory (POP-1), Mr. Richard Quigley, Project Engineer.

This report was submitted by the author in November 1970.

Publication of this technical report does not constitute Air Force approval of the report's findings or conclusions. It is published only for the exchange and stimulation of ideas.

Glen M. Kevern
Chief, Energy Conversion Branch
Aerospace Power Division

ABSTRACT

The design, fabrication, test and evaluation of six foot nominal diameter multicycling metallic diaphragms for the storage and positive expulsion of approximately 500 pounds of liquid hydrogen from a spherical tank are described. Diaphragm fabrication methods suitable for this large size were studied and developed. Weight trade-off studies for subcritical and supercritical diaphragm/tank expulsion systems for cryogenic hydrogen and oxygen were conducted.

The positive expulsion diaphragm demonstrated consists of a thin, one-piece stainless steel hemispherical type shell which is reinforced by stainless steel hoop wires attached by brazing. Diaphragm performance was verified by successful reversal testing. A technique for forming large, one-piece thin metal shells to precise contour and close thickness tolerance was demonstrated. Use of copper plating to apply braze material to parts joined by furnace brazing was proven.

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1. INTRODUCTION

There is a need for reliable and lightweight cryogenic storage and positive expulsion systems for a variety of missions. Subcritical cryogenic systems with their low storage and operating pressures, offer the potential for reliable operation with significant weight savings, primarily due to reduced storage and expulsion tankage weight. A key technology area which must be demonstrated in order to realize these potential advantages of subcritical systems is a reliable positive expulsion diaphragm for cryogenic service. Development of suitable non-metallic or combined plastic-metallic diaphragms has met with indifferent success due to reliability, compatibility and porosity problems, as well as other factors. Metallic diaphragms offer a solution to these problems.

Reliable, lightweight, multicycle capability, metallic, positive-expulsion diaphragms have been demonstrated in size ranges from six inches to thirty-three inches in diameter for cryogenic and other fluid service [(1)-(7)]. Present and contemplated missions require fluid storage and positive expulsion devices in size ranges from four to six feet diameter and larger. Demonstration of a reliable metallic diaphragm in this large size range is, therefore, needed to meet these requirements. This report describes the work performed by Arde, Inc. under Contract No. AF 33(615)-2827, in demonstrating a six foot nominal diameter multicycling metallic bladder sized for the storage and positive expulsion of approximately 500 pounds of liquid hydrogen from a spherical tank. The work reported herein is an outgrowth of previous effort performed by Arde for the Air Force under Contract No. AF 33(657)-11314 which culminated in the successful demonstration of a multicycling metallic diaphragm for cryogenic service in a 23" nominal diameter size.

Program effort under the present contract consisted primarily of the design, fabrication, test and evaluation of the six foot diameter diaphragm. In support of this work, diaphragm fabrication techniques were studied, developed and evaluated by tests. Appropriate special tooling, test specimens and test rigs were designed, fabricated and utilized. In addition, a study of sub-critical and supercritical diaphragm/tank expulsion systems for cryogenic service was performed to provide initial guidelines for system weight trade-offs.

2. SUMMARY OF RESULTS

2.1 The design and fabrication of a six foot diameter modified hemispherical stainless steel ring reinforced multicycling positive expulsion diaphragm was verified by reversal testing using room temperature water as the pressurant. Four (4) complete reversals were successfully accomplished. Diaphragm reversal modes and actuation pressure levels were in accordance with design predictions.

Diaphragm deflections were well controlled with the six foot diaphragm rolling through each reinforcing ring, one at a time in sequence, as desired, until the diaphragm was completely reversed. Actuation pressures (pressure differences across the diaphragm during reversal) varied from 1 to 4.5 psid from start to reversal completion.

2.2 Diaphragm fabrication techniques suitable for large size were successfully demonstrated in six foot diameter hardware. Subscale efforts were used to verify the fabrication approaches.

2.2.1 A diaphragm shell forming technique for constructing large, thin, prescribed thickness and contour shells to close tolerances was developed. This technique, which started with a tapered thickness flat sheet, was based on the use of hydraulic forming coupled with test - verified plasticity theory. Thickness and contour control demonstrated in nominal six foot diameter size and 25 mil wall thickness was ± .06' on diameter and ± 1.5 mils on thickness. Diaphragm shell material was AISI 321 stainless steel.

2.2.2 The use of copper plating to apply braze material to parts joined by furnace brazing was demonstrated. Copper plated 5/16" Ø AISI 308 stainless steel rings were successfully attached to the six foot diameter 25 mil thick 321 stainless steel diaphragm shell by means of furnace brazing.

2.3 A weight trade-off study for subcritical and supercritical cryogenic fluid storage and expulsion systems was performed for a prescribed zero g mission. The results showed a substantial weight advantage for subcritical systems for cryogenic hydrogen and oxygen.

3. DESCRIPTION OF METALLIC DIAPHRAGM

The positive expulsion diaphragm (herein sometimes called bladder) consists of a thin, one-piece stainless steel hemispherical type shell which is reinforced by stainless steel hoop wires (Figure 1). The hoop reinforcement is attached to the shell by means of brazing. The function of the hoop reinforcement is to prevent random buckling and to control the diaphragm rolling mode during reversal and fluid expulsion. Although developed for cryogenic fluids, the diaphragm materials utilized (stainless steel and copper or gold brazing) make it also suitable for use with a wide range of storable propellants. When such a hoop reinforced diaphragm is housed inside a tank (Figure 2) and fluid is stored on the concave side of the diaphragm, the application of a pressure on the convex side of the diaphragm that exceeds the fluid pressure by a few psi will cause the diaphragm to invert at its apex and pass through successive positions (e.g. positions 1-6 in Figure 2) until the diaphragm is completely inverted and the fluid is expelled. Controlled rim-rolling deformation modes, wherein the diaphragm begins to invert at its rim, and combination rim-and-apex-rolling modes can also be achieved.

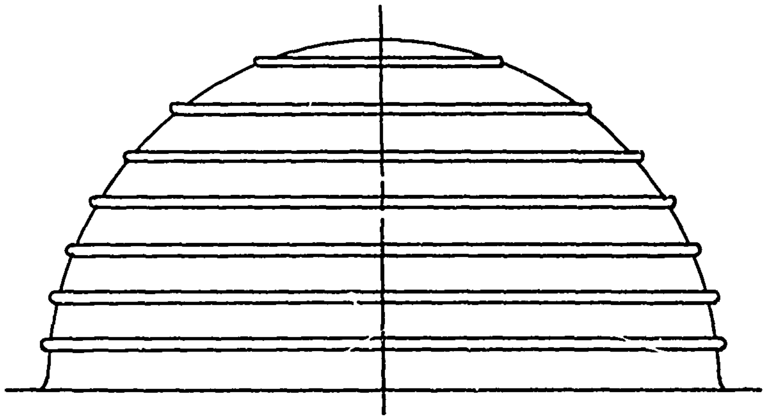


FIGURE 1. METALLIC POSITIVE EXPULSION DIAPHRAGM

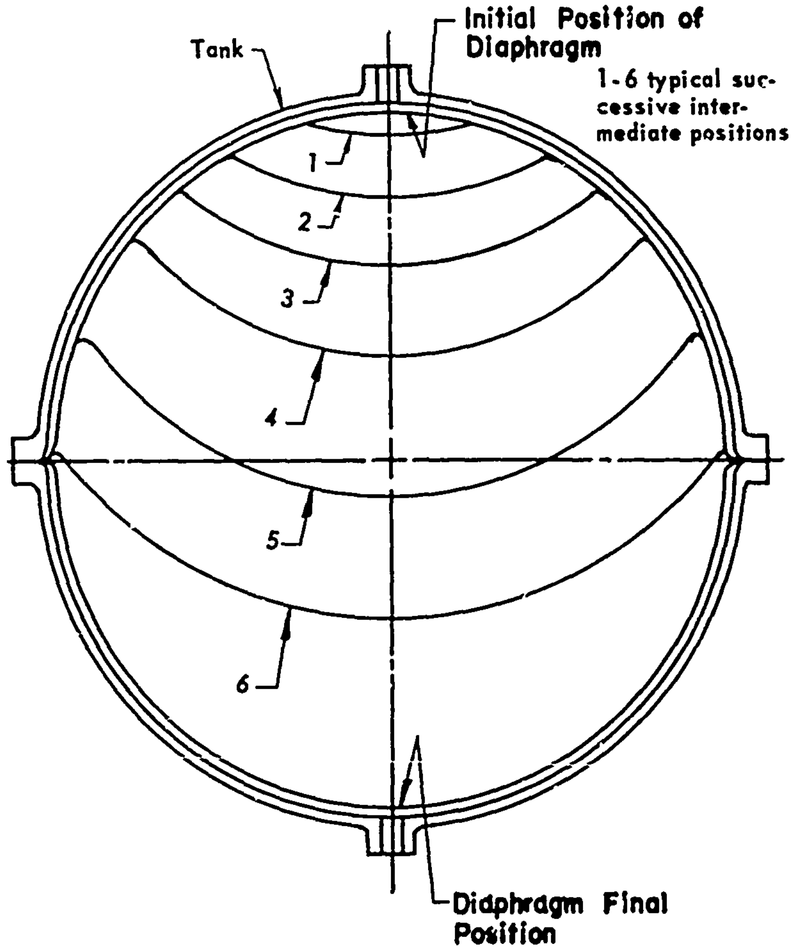


FIGURE 2. DIAPHRAGM REVERSAL

4. DIAPHRAGM STRUCTURAL DESIGN CONSIDERATIONS

Structural design of a hemispherical ring reinforced diaphragm with a given diameter involves selection of shell thickness and contour and reinforcing wire size and spacing as well as wire to shell joint configuration. The six foot diameter diaphragm configuration used in the program was a scale up of the 23" multicycle capability diaphragm previously demonstrated by Arde under Contract AF 33 (657)-11314, references 1 and 2, Figures 3 and 4 define the six foot diaphragm configuration. A 25 mil thick shell and 5/16" diameter circular cross-section reinforcing wire with 1.6" spacing was employed as shown.

The shell is made as thin as possible to reduce bending strain and to lower the diaphragm actuation pressure during diaphragm reversal. The size and spacing of the wire reinforcement (stiffeners) is selected to control the bladder deformation mode and preclude random buckling. A satisfactory design is one which exhibits lower actuation pressures than critical buckling pressures for the complete diaphragm reversal cycle. The actuation pressures are the pressure differences across the diaphragm required to roll the shell structure through each wire in a controlled deformation mode similar to the one sketched on Figure 2. The critical buckling pressures are the lower of the pressures required to produce either overall or local compressive instability of the stiffened shell structure. Reinforcing wire interference during diaphragm deflection has to be avoided. In addition, the wires should be spaced far enough apart to permit one wire at a time to roll without affecting the other wires.

A conical transition region is used at the diaphragm equator to avoid "theoretically infinite" acutation pressure there, and in addition, to reduce the diaphragm bending strain since the total angle turned through during bladder reversal is decreased. Diaphragm reversal cycle life is therefore increased as the cone angle is made larger.

As is often the case, trade-offs have to be made between conflicting requirements. For low shell bending strain and actuation pressure, the shell should have a small thickness and the reinforcing wires have a small cross-section and be spaced far apart. This reduces the diaphragm buckling resistance. For increased buckling resistance, shell thickness and wire cross-section need to be increased and wire spacing reduced. Further compromise is required when the wire spacing needed to preclude buckling for a given wire size and shell thickness is small enough to lead to wire interference during diaphragm reversal. Finally, use of larger transition cone angles for increased reversal

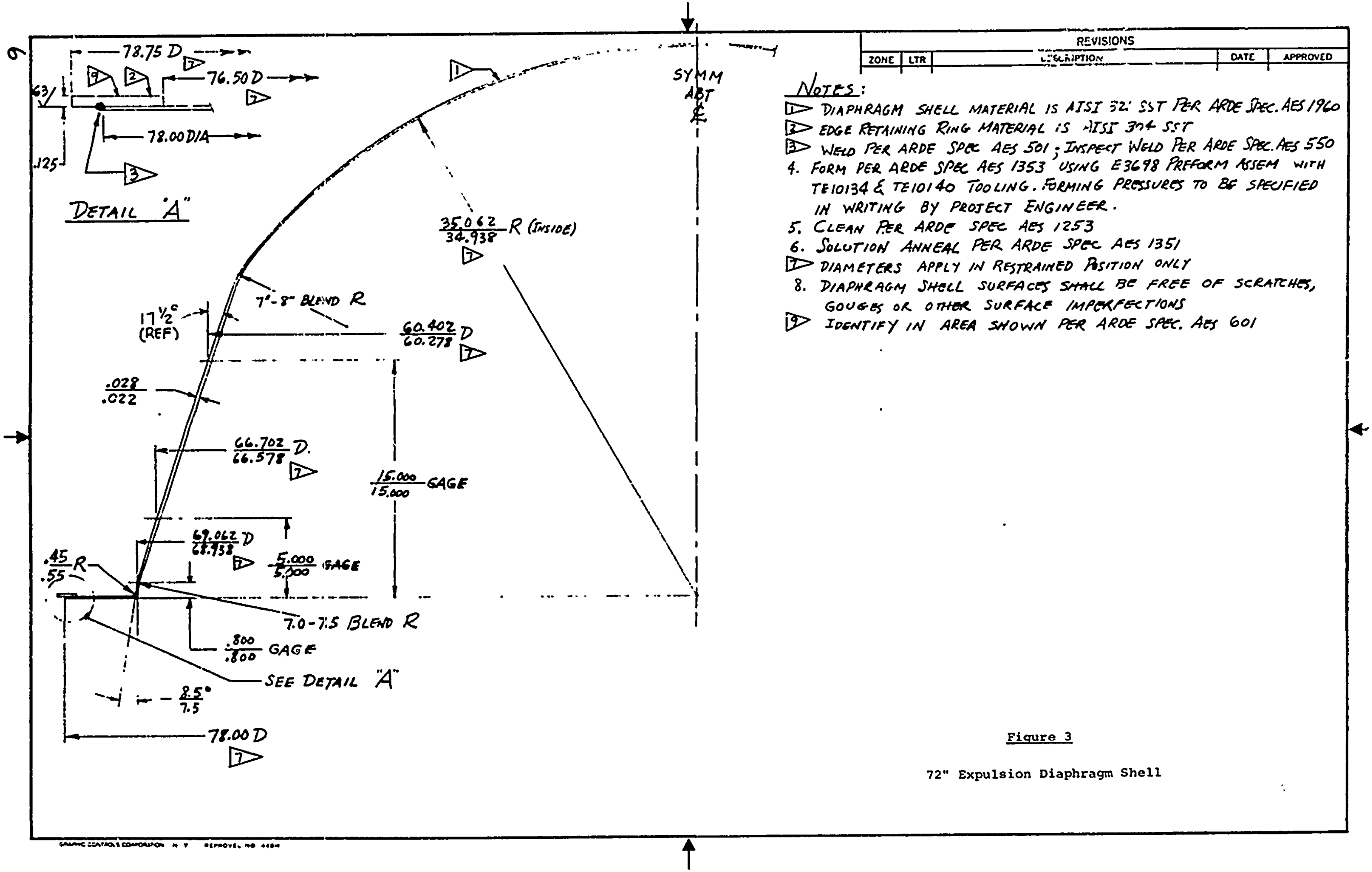


Figure 3

72" Expulsion Diaphragm Shell

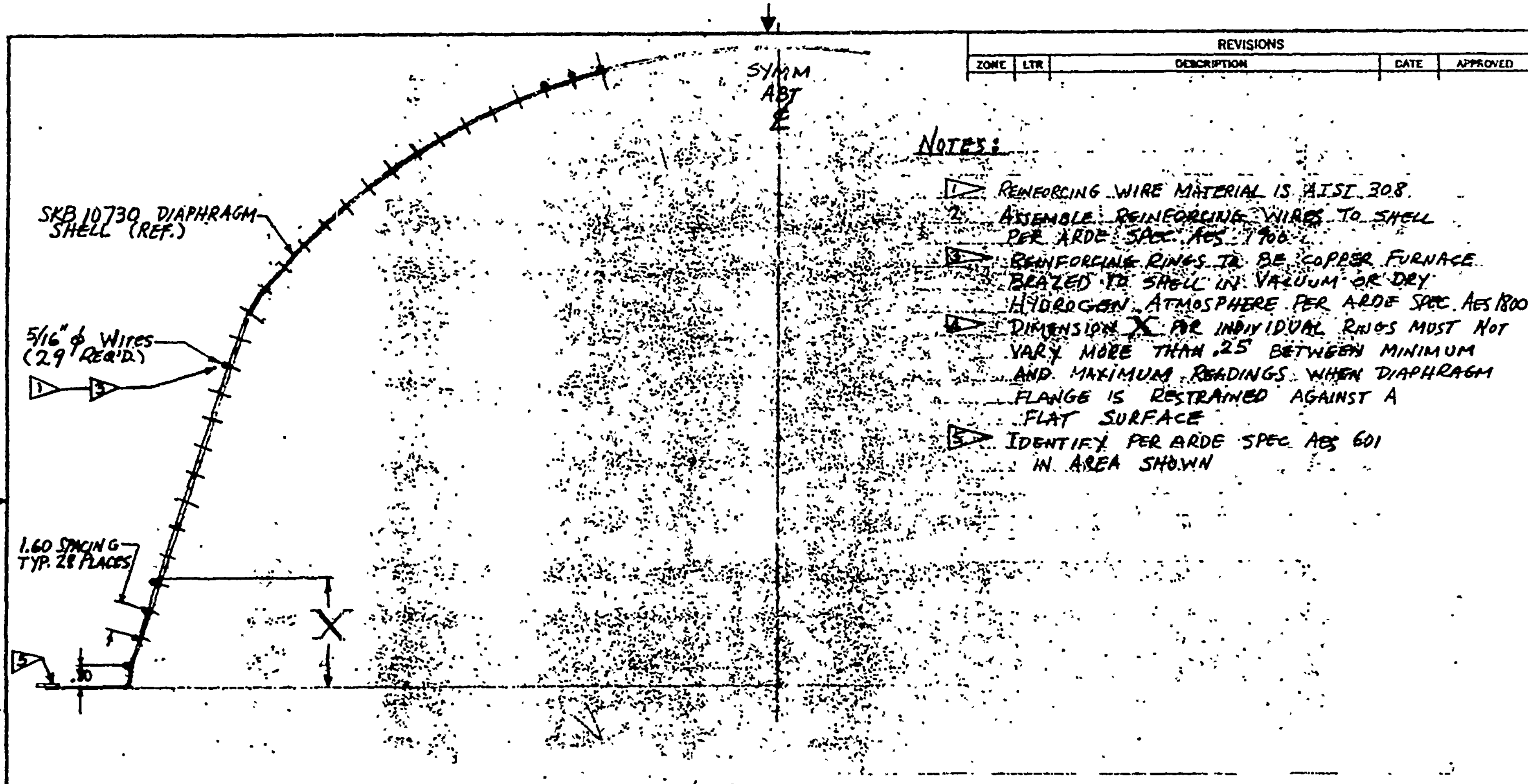


Figure 4

72" Expulsion Diaphragm Assembly

cycle life has to be tempered by reduced "packaging efficiency" when the diaphragm is housed in a tank to form a complete expulsion tank assembly. Contouring the tank to correspond with diaphragm shape avoids this penalty.

The stiffener ring to shell brazed joint configuration is designed to produce minimum strain concentration during diaphragm reversal. To accomplish this, the braze meniscus is made as small as possible and the brazing parameters chosen to provide a ductile joint to accommodate the large strains imposed by diaphragm deformation. These aspects are discussed in somewhat more detail in sections 5 and 6 which follow.

5. DIAPHRAGM MATERIAL SELECTION

The materials used for the six foot diameter diaphragm were the same as those employed in the construction of the 23" diameter multicycle diaphragm demonstrated in a previous program. The criteria used for material selection were:

- Large elongation to necking capacity with relatively low work hardening at cryogenic temperatures.

- Compatibility with contained fluids and pressurants (LH_2, GH_2 and GHe).

- Reinforcing wires readily attached to shell with strong and ductile brazed joint.

- Capable of being readily formed into rings and thin shells.

- Compatibility with tank material with minimum dissimilar metals problems and good weldability. Diaphragm is welded into tank in flight type cryogenic storage and expulsion system.

AISI 321 annealed stainless steel was chosen for the diaphragm shell and annealed AISI 308 stainless steel wires (standard weld wire) were used in the construction of the reinforcing rings. Copper was used as the braze material to join the rings to the diaphragm shell. This material was applied to the wire in the form of thin copper plating. The brazing parameters were chosen to produce a small braze meniscus and to minimize the diffusion of braze into the parent material as detailed in Section 6.

6. DIAPHRAGM FABRICATION

6.1 General Considerations

The primary problem in the six foot diameter diaphragm program was the development of fabrication techniques suitable for the large size since diaphragm design theory was previously verified in numerous programs. The most critical fabrication problem was the construction of the thin, one-piece prescribed close tolerance thickness and shape diaphragm shell. No satisfactory fabrication techniques apparently existed for this size and type of diaphragm shell prior to program completion. The shell forming methods demonstrated in the present program utilized previously developed Arde technology in the plastic deformation of metals.

6.2 Diaphragm Shell Fabrication

The diaphragm shell fabrication technique successfully developed is basically a hydraulic forming process wherein the starting sheet material is formed into the final prescribed shell shape and thickness by hydraulic pressure after a succession of forming passes and intermediate anneals. The precise final shape is achieved by use of a sizing die. Several important differences between this process and conventional forming techniques contributed to the success achieved. First, the edge of the sheet (and subsequent shell) is restrained to be at a specified diameter throughout the forming process and thus the boundary conditions at the edge are always known. This eliminates problems due to clamping pressure, sealing, friction, thin-out, etc., common to those fabrication methods which allow the edge to move inwards during forming. Second, the starting sheet has a prescribed tapered thickness variation. This prevents thin-out and overstraining and permits close control of shell thickness and shape. The starting sheet taper thickness variation (determined by means of test-verified plasticity theory) is selected to give the prescribed final shell thickness and contour after a specified number of forming passes and intermediate anneals. Finally, the tooling is simple and not particularly sensitive to size increases. This eliminates press capacity or other tooling size of capacity problems and leads to reduced fabrication cost.

Full scale 72" diaphragm shell forming was preceded by subscale effort. The subscale work in 7" and 20" sizes served to verify the plasticity relations used and to check out the tooling concepts and fabrication processing.

6.2.1 Subscale Diaphragm Shell Forming

6.2.1.1 7" Shell Forming

The initial 7" diameter effort utilized existing tooling used by Arde for the forming of spherical segment burst diaphragm shells for very high pressure shock tubes. Two flat sheets of annealed AISI 321 stainless steel were clamped between two 7" I.D. forming rings and an intermediate spacer ring. Hydraulic pressure was then applied between the sheets to "bulge form" them into curved surfaces. Bolts and "O" ring seals were used to contain the pressure and hold the sheet edges. Three (3) deformation-anneal cycles were required to form the hemispheres. Both constant thickness and tapered thickness sheets were employed in this effort. The apex of the hemispheres formed from the constant thickness starting sheets thinned down 300% compared to the equator region thickness. The starting sheet taper was desigend to reduce the thin out and approach uniform shell thickness in the finished hemisphere. The first trial sheet taper design (sheet edge 70% of center thickness) reduced the thin out in the finished formed 7" I.D. hemisphere to 23%. Figure 5 shows 7" I.D. hemispherical shells formed from constant and tapered starting flat sheets.

6.2.1.2 20" Shell Forming

Twenty inch (20") nominal diameter shell forming was next performed utilizing tooling configurations, starting sheet preform material and taper thicknesses as well as edge retaining ring configurations projected for use on the full scale 72" diaphragm shell. An edge retaining ring welded to the starting tapered circular sheet was used for handling purposes, edge retention and fixity during forming and as part of the shell furnace support system during annealing and subsequent brazing when reinforcing rings are attached to the shell to form the finished diaphragm assembly.

The starting tapered thickness circular sheet with its welded on edge retaining ring was clamped by means of bolts between the flange of a pressure closure and a 20" I.D. forming ring. O' rings were used as seals. Hydraulic pressure was applied to plastically "bulge form" the sheet into a

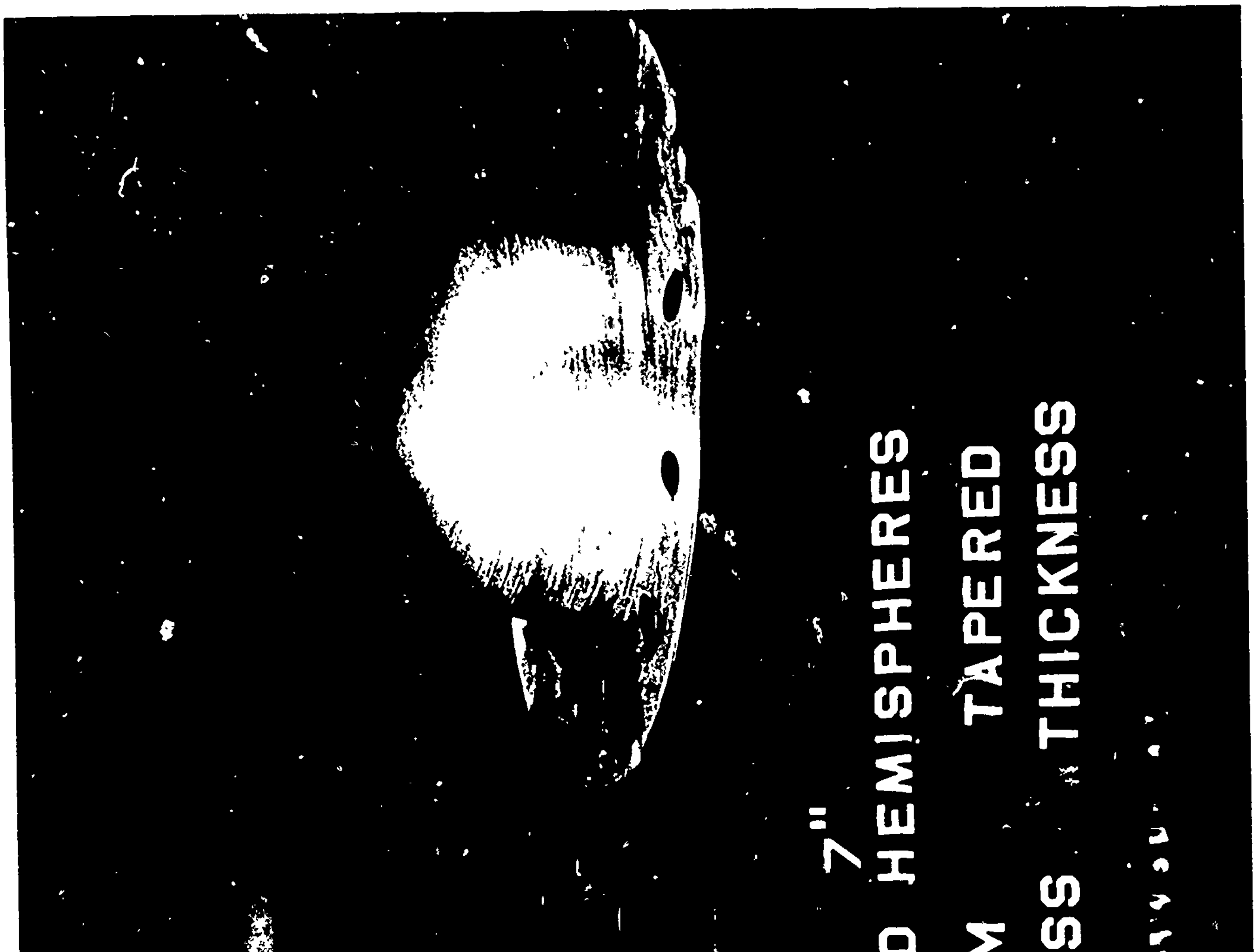

7" FORMED HEMISPHERES
UNIFORM THICKNESS
TAPERED THICKNESS

FIGURE 5

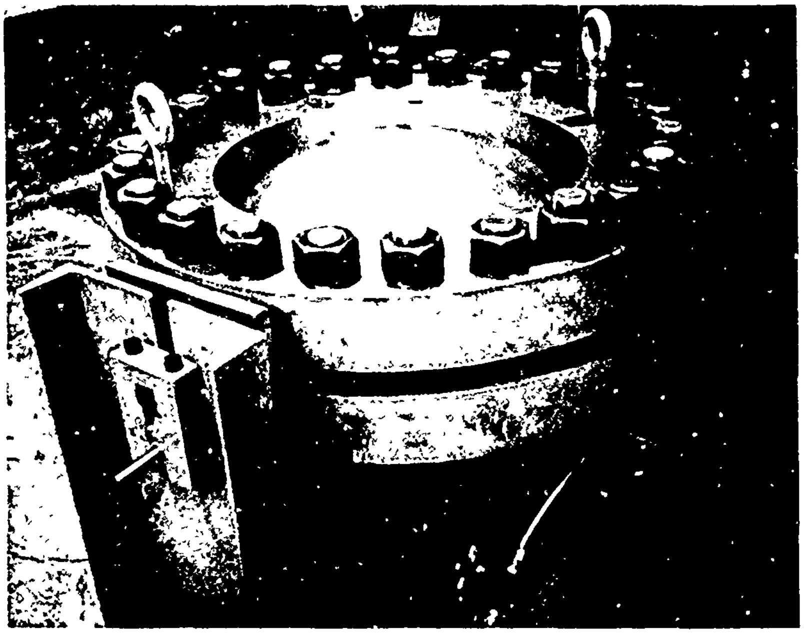
hemispherical shape. Four (4) passes with intermediate anneals were used to form the hemispherical shape. Figure 6 shows the shell during and after the second hydraulic forming pass. The forming ring was then replaced by a 23" I.D. 15° conical forming die for final forming of a 23" I.D. 15° cone angle modified hemispherical diaphragm shell. Two (2) more passes and intermediate anneals were used to final form the 23" diameter modified hemispherical diaphragm shell. The completed shell is shown on Figure 7.

The successful 23" subscale shell forming proved out the plasticity design, checked scaling from 7" size and verified the tooling and fabrication processing. Scale up from 23" to 72" size using the same material, taper sheet thickness, tooling configuration and fabrication techniques was, therefore, made with confidence.

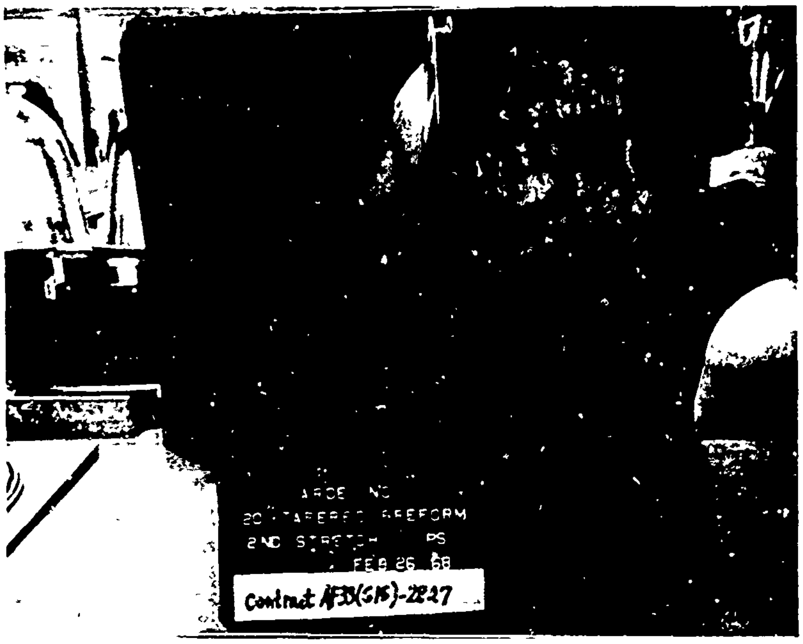
6.2.2 Full Scale Six Foot Diaphragm Shell Forming

Six foot diameter diaphragm shell fabrication started with procurement of 96" x 96" x .05" thick sandwich pack rolled and annealed 321 stainless steel sheet followed by taper grinding to prescribed thickness variation. Material manufacture and sheet rolling was performed by U. S. Steel Corporation. Sheet taper grinding was done by Mill Polishing Corporation, Delair, New Jersey. Figure 8 shows the finished tapered thickness sheet being inspected at Arde. A two-step taper was used. The outer edge was maintained at 30 mils thickness.

Edge retaining rings were next welded to the trimmed, tapered thickness circular starting sheet. The tapered sheet was then clamped between a forming ring and the flange of a head closure and hydraulically formed (bulged) into a shell "preform" stage using intermediate anneals between forming passes. The shell preform was then final sized, by hydraulic pressure in the final sizing die using several passes and intermediate anneals. The final sizing die consisted of a removable plastic liner mounted in the water reversal test rig which served as the liner structural support. The liner was molded directly into the reversal test rig and then machined to final inside contour. A flanged hemispherical head closure used for clamping the



Shell in Forming Rig During Hydraulic Stretch



Shell After Second Hydraulic Stretch

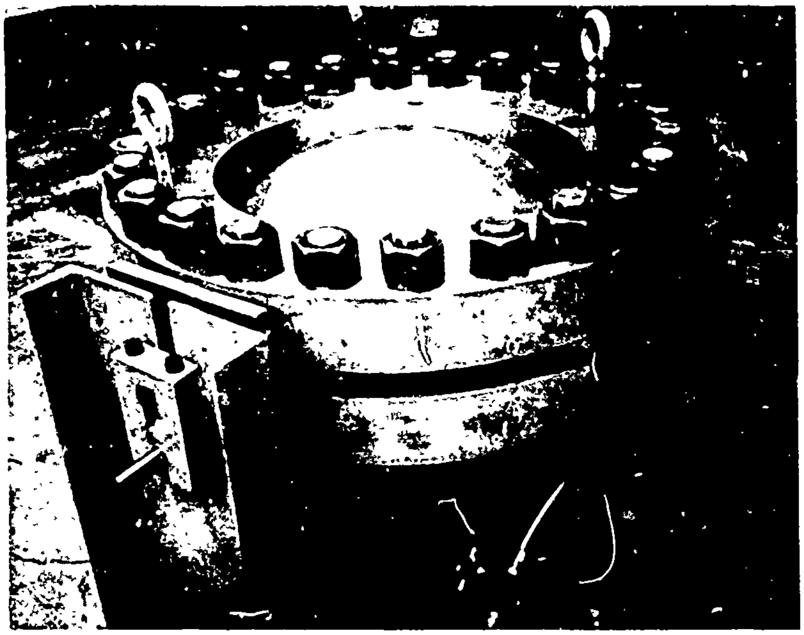
20" SUBSCALE BLADDER SHELL FORMING

FIGURE 6

ARDE, INC.

23" BULGE FORMED PRESCRIBED THICKNESS BLADDER SHELL

FIGURE 7

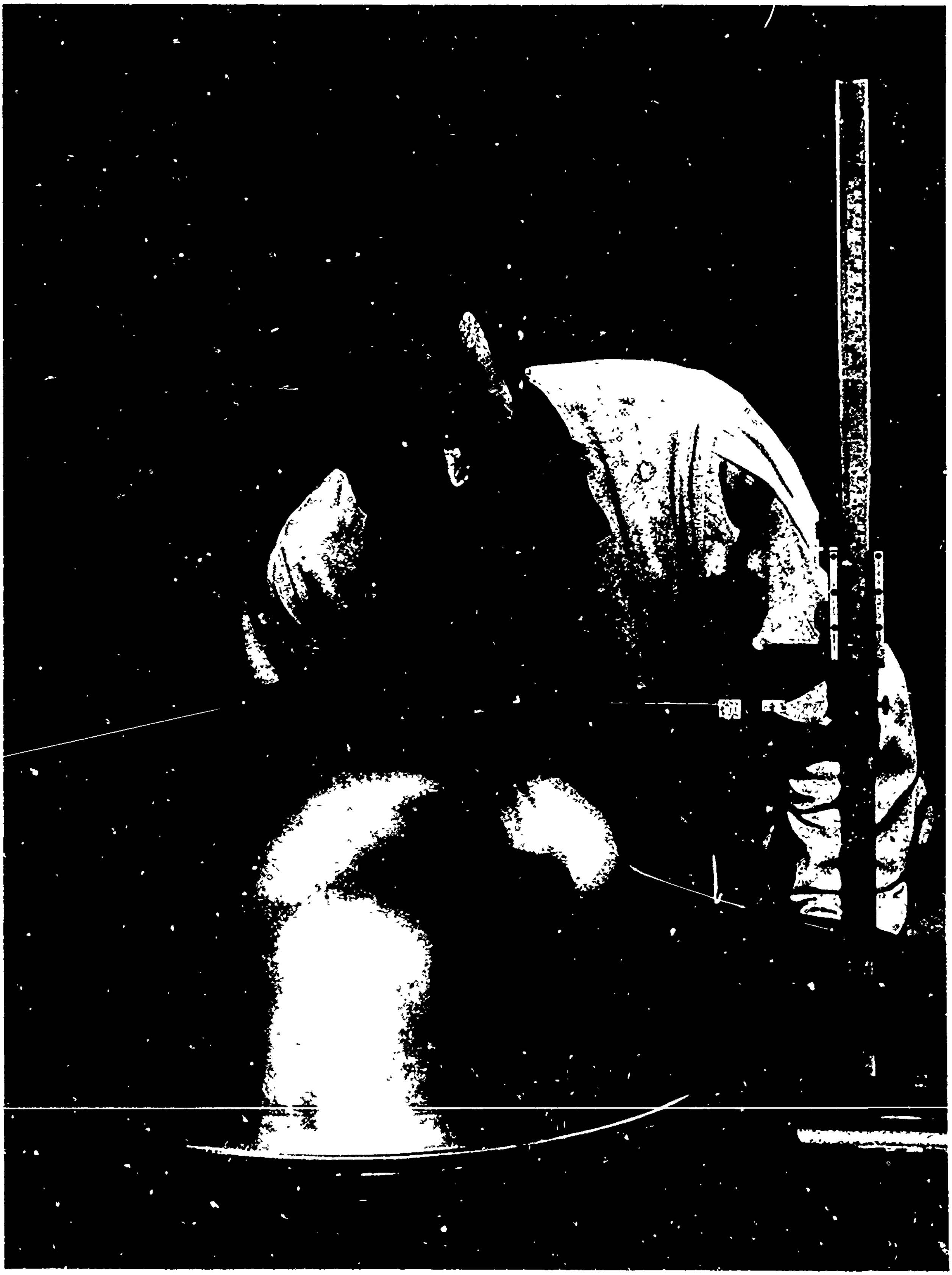


Shell in Forming Rig During Hydraulic Stretch

Shell After Second Hydraulic Stretch

<u>20" SUBSCALE BLADDER SHELL FORMING</u>

FIGURE 6



ARDE, INC.

23" BULGE FORMED PRESCRIBED THICKNESS BLADDER SHELL

FIGURE 7

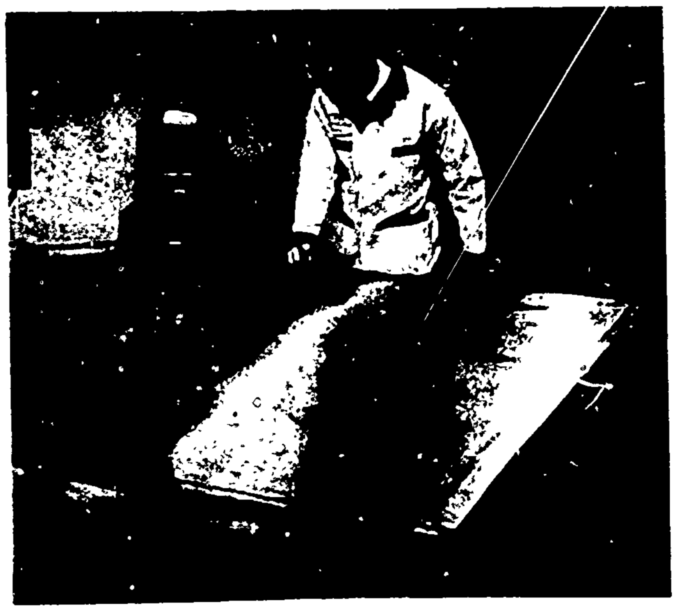


FIGURE 8. INSPECTION OF TAPERED THICKNESS SHEET

shell preform in the forming tool and permitting the die to be pressurized, completed the final sizing die configuration.

Figure 9 shows several sequences in the shell fabrication process. The final formed six foot modified hemispherical diaphragm shell is shown in Figure 10. The 17 1/2° conical transition region in the shell equator area is used to minimize strain and reinforcing wire interference problems during diaphragm reversal. The shell thickness was controlled at 25 ± 1.5 mils (vs. 25 mils nominal target) while the six foot nominal diameter shell shape was well within the ± .06" diameter tolerance band target.

6.3 Six Foot Diameter Diaphragm Assembly Fabrication

6.3.1 Definition of Fabrication Techniques

The diaphragm assembly consists of hoop reinforcing wires attached to the diaphragm shell as shown in Figure 3. The reinforcing wires are attached to the shell by furnace brazing. Heretofore for smaller sized diaphragms, the copper braze material had been applied in a paste or wire form and the wires were tack welded to the shell to hold them in place during brazing. In the search for fabrication methods suitable for the large size, brazing technique investigations were made . For better control and ease of fabrication, it appeared desirable to apply the braze material as copper plate on the formed reinforcing wires prior to fitting and attaching them to the shell. Use of brazing fixtures wherein the shell and wires were clamped together in the furnace during brazing, in order to eliminate tack welding the rings to the shell, was also investigated.

6.3.1.1 Brazing Tests Using Copper Plated Braze Material on Reinforcing Wires

Subscale tests were made to define and verify the brazing of reinforcing wires to diaphragm shell using copper plate as the braze material.

a) 6" Diaphragm Tests

Six inch (6") diaphragm shell specimens with 1/8", 3/32" and 5/32" Ø reinforcing wires copper plated with plating thicknesses 1 to 3 mils thick were fabricated and brazed in dry GH_2 and vacuum atmospheres. Various gaps between the wires and shells (1 to 6 mils) were used to investigate the range of fit up required to obtain satisfactory copper brazed reinforcing wire to diaphragm shell joints. Some of the brazed 6" diaphragm specimens were cut up to make pull test specimens

BULGE FORMING PROCESS STEPS

a. Welding edge retaining ring to 78" dia. tapered sheet.

b. Free form bulging; first pass.

c. Inspection of part after first pass.

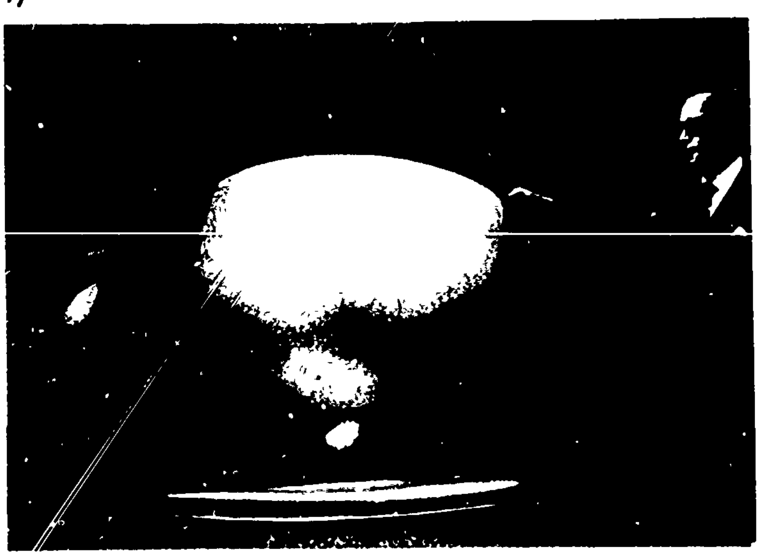
d. Free form bulged 63" dia. hemisphere prior to final sizing operations.

e. Assembly of bulged shell into final sizing die.

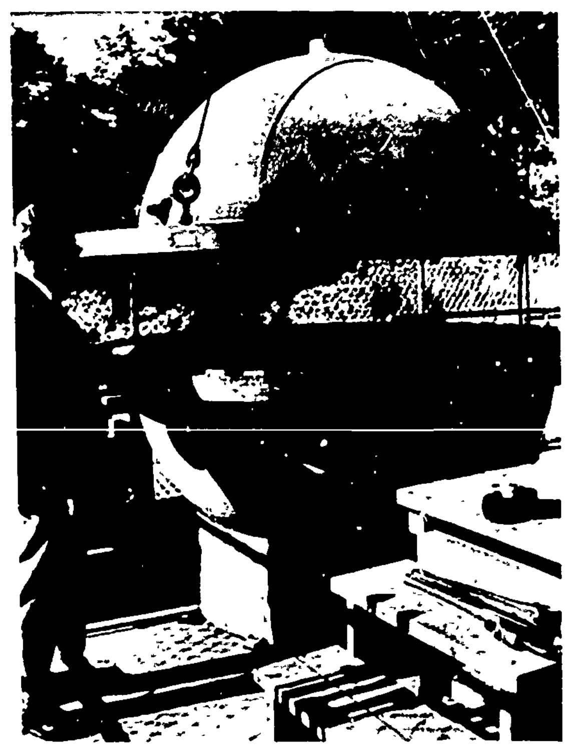
a)

b)

c)



d)



e)

Figure 9

FIGURE 10. 72" DIAMETER CONTROLLED THICKNESS DIAPHRAGM SHELL AFTER FINAL FORMING

and then sectioned and examined under high magnification. Other 6" diaphragm braze specimens were reversal tested in an existing 6" reversal test rig. These tests indicated that the use of copper braze material in the form of plating on the reinforcing wires was a feasible approach. The brazed joint was found to be stronger than the parent material. Copper plating thicknesses as small as 1 mil and fit-up gaps up to 6 mils produced good brazed joints. As a rule, the smaller the fit-up gap, the better the overall joint appeared. For a small and controlled braze joint meniscus, fit-up gap should be held to about 3-4 mils maximum, which is much greater than the 1 to 1.5 mils maximum recommended by the brazing vendors for copper. Figure 11 shows a 6" diaphragm specimen braze pull test specimens and a 25X magnified braze joint cross-section. Braze porosity and some diffusion of copper into the parent material, evident from the magnified cross-section view, can be eliminated by proper brazing time-temperature variation and use of less braze material. Even though this joint had a three mil gap and excess braze material (3 mil thick copper plate) was used, the meniscus control and extent of braze coverage are good.

b) <u>12" Subscale Brazing Models</u>

Two (2) subscale 12" diameter brazing models 25 mils thick were fabricated. Copper plated reinforcing wires (5/16" Ø) with plating thicknesses 1 to 5 mils were tack welded to the cylinders. The materials, thickness and reinforcing ring cross-section diameters used were identical to those projected for the full scale 72" diaphragm. The models were furnace brazed in vacuum and dry GHe atmospheres to determine the "best" copper plating thickness to be used for the 72" diaphragm assembly. Based on these tests, a two (2) mil thick copper plate was selected.

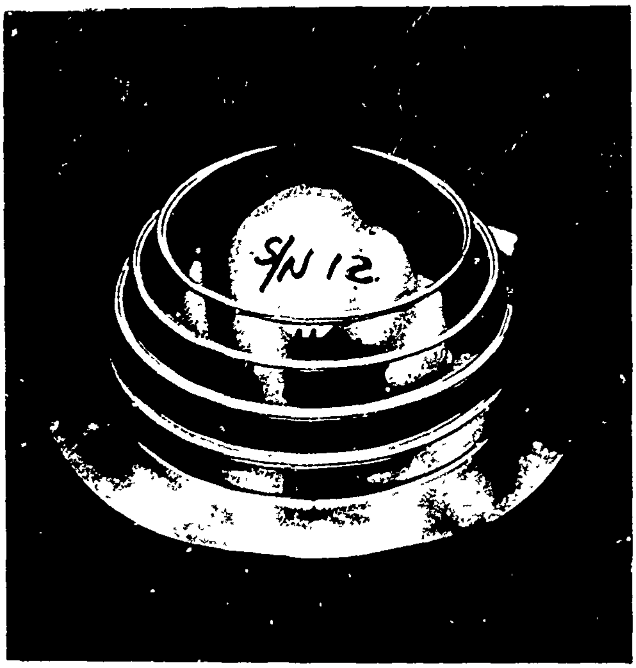
c) <u>Brazing Fixture Model Tests Using Copper Plated Wires</u>

Flat plate models simulating brazing fixture concepts projected for the 72" diaphragm were fabricated and tested. The objective of this work was to eliminate the use of the tack welds which hold the reinforcing wires to the shell during brazing. In one concept which appeared promising, the diaphragm shell would be clamped between two "rigid" shells. Reinforcing wire supports (consisting of plated and unplated wires)

72" BLADDER PROJECT

Copper Plated Wire Brazing Study

6" Subscale Specimens



6" Bladder Specimen with Copper Plated Wires after Brazing

Braze Pull Test Specimens

Cross-Section of Braze Specimen 25x Magnification

FIGURE 11

bearing on the rigid shells and the diaphragm shell would be used to locate the reinforcing rings and to apply the contact pressures needed for brazing. Stop off would be used to inhibit brazing components together where not desired. Flat plate brazing model tests were successful and showed feasibility of concept. Figure 12 shows views of a flat plate brazing fixture model before and after successful brazing.

Despite initial feasibility indications, however, the brazing fixture approach was not used in the fabrication of the full scale 72" diaphragm. The reinforcing rings were tack welded to the diaphragm shell prior to brazing as was done in all previous programs. The reasons for this were: 1) the cost in time and dollars to demonstrate feasibility using actual shell type fixtures and 2) the relatively few full scale parts required.

6.3.2 Six Foot Diaphragm Assembly Fabrication Procedures

The six foot diaphragm assembly was fabricated as described below using the results of the brazing investigations detailed in the preceding sections.

Following shell forming, the reinforcing wires were fabricated and attached to the shell by tack welding. The wires were then permanently attached to the shell by furnace brazing, the tack welds functioning as the brazing fixture. The brazing material was applied as copper plating on the wires prior to their tacking to the shell. This procedure offers the potential for significant improvement in brazing technology for diaphragm construction since the braze material is applied in a controlled manner with minimum labor. Brazing was accomplished by Wall Colmonoy Corporation utilizing one of their 100" inside working diameter vacuum furnaces. The same furnace was also used to anneal the diaphragm shells during forming. Figures 13 and 14 show the wires being tack welded to the shell and the diaphragm inserted into the vacuum furnace prior to brazing. A completed six foot diaphragm assembly is shown in Figure 15.

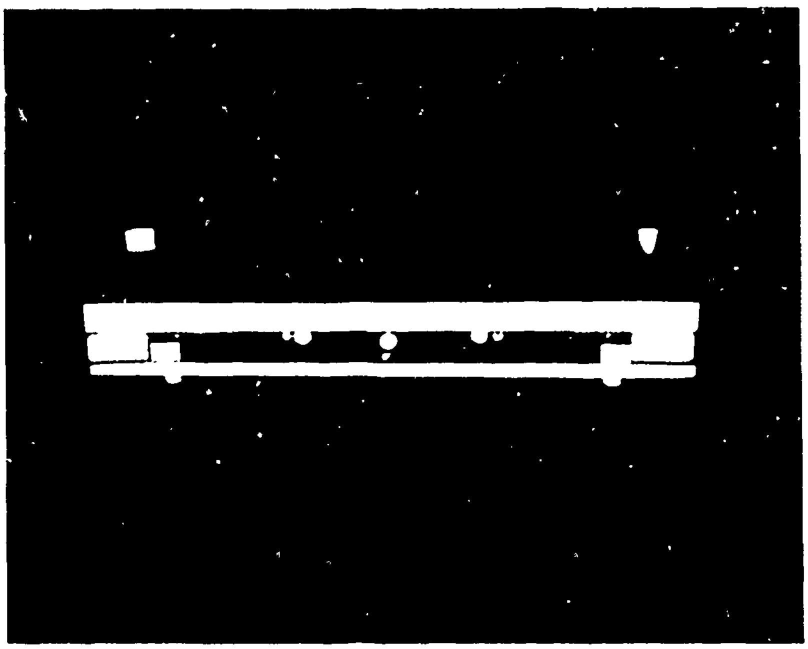
Inspection of the completed diaphragm assembly at Arde after brazing revealed brazing voids in several wires and shell leaks at three tack weld regions. The brazing voids were repaired by silver solder and the shell leaks were weld or silver solder repaired utilizing repair techniques developed and verified by Arde in the previous 23" diaphragm

ARDÉ, INC.

72" BLADDER

FLAT PLATE BRAZING

FIXTURE MODEL



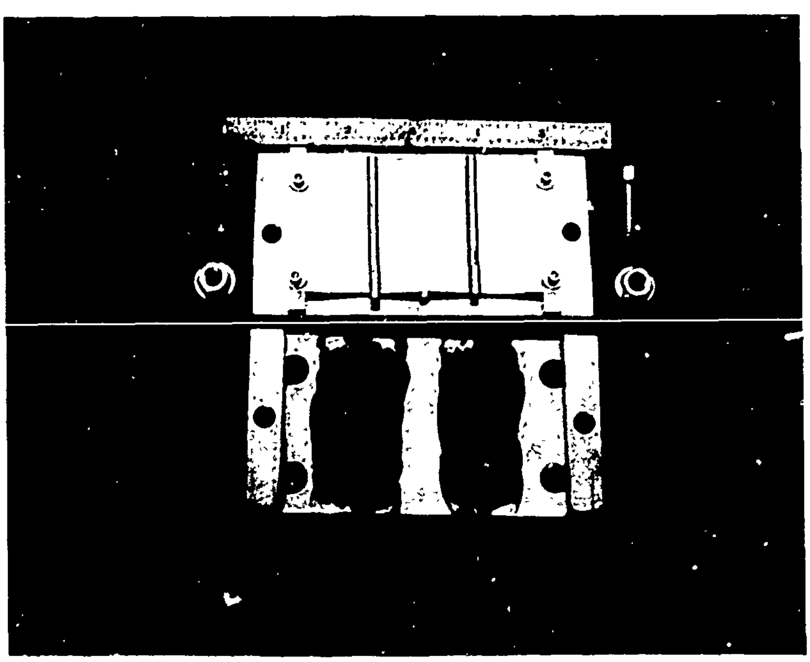
End View Showing:

a) Small unplated wires (upper outer wires) used for locating larger Cu plated wires (upper inner wires).

b) Lower unplated wire used as support.

c) Sheet to which two (2) upper Cu plated wires are to be brazed.

d) Bolts and load plates which apply clamping forces to sheet through inner three (3) larger wires to force fit up for brazing. Braze stop-off used on mating surfaces of Cu plated wires with upper plate and positioning wires.



Exploded View Showing:

a) Two (2) Cu plated wires brazed to sheet. Cu plate used as braze material.

b) Black braze Stop-off material on positioning wires and flat plate.

FIGURE 12

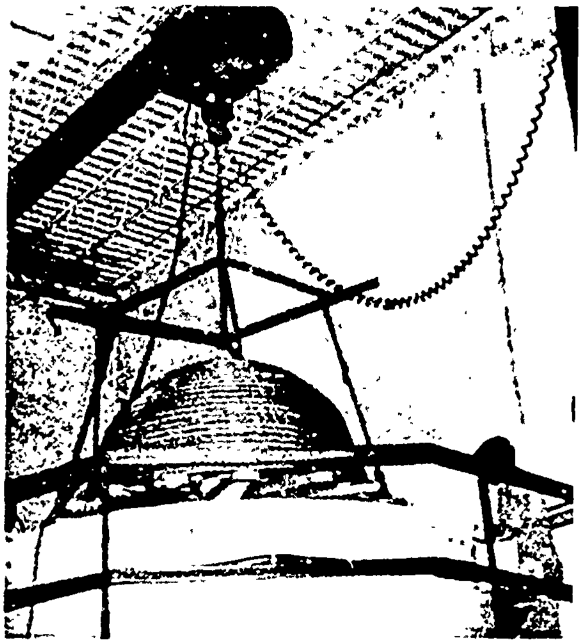


FIGURE 14. INSERTION OF 6 FT. DIAMETER DIAPHRAGM INTO VACUUM BRAZING FURNACE

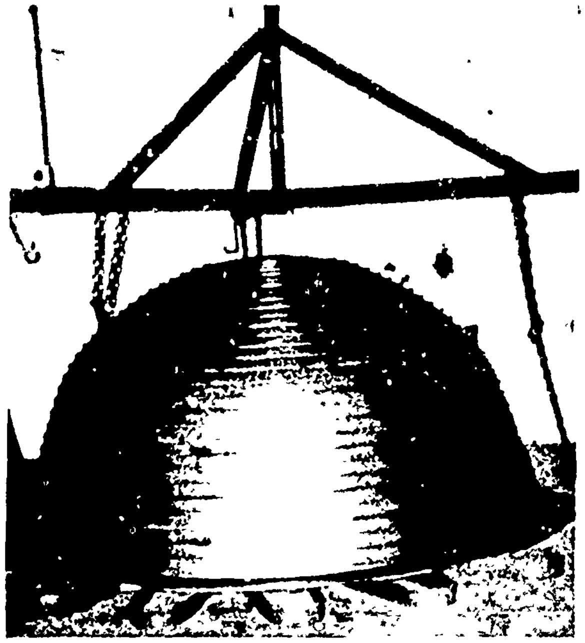


FIGURE 15. 72" DIAPHRAGM ASSEMBLY PRIOR TO BRAZING

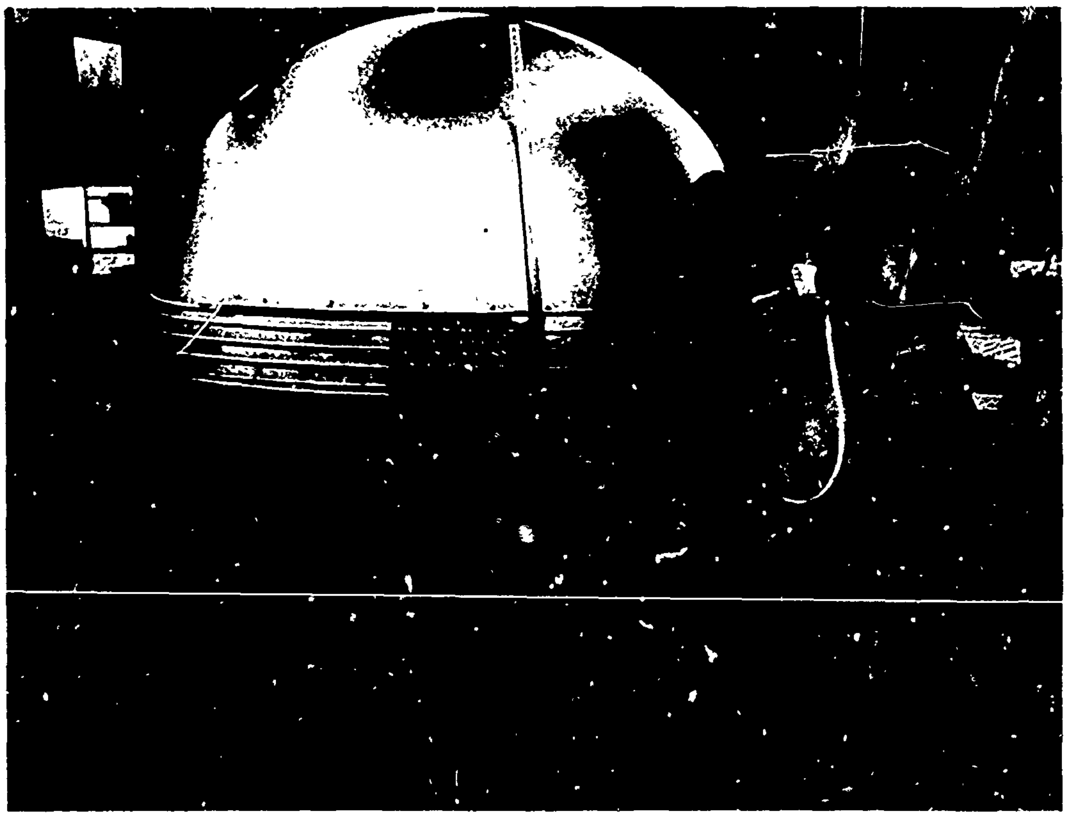


FIGURE 13. TACK WELDING WIRES TO 72" DIAPHRAGM SHELL

program, references 1, 2.

The number of defects were relatively few considering that this was the first diaphragm of such a large size to be fabricated and that hand assembly techniques with minimal tooling were used.

The fabrication problems were brought about primarily by the relative stiffness of the 5/16" Ø wire compared to the large diameter thin shell making fit up and application of proper clamping pressure needed for tack welding difficult. It is anticipated that these problems will be eliminated through improved fabrication processing and tooling as was done for the smaller diaphragms (up to 33") successfully built by Arde and verified by test.

7. DIAPHRAGM TESTING

Six foot diaphragm design and construction were verified by reversal testing in a reversal test rig utilizing water as the pressurant. The diaphragm is clamped between the flanges of a hemispherical closure and loose circular rings by bolts. "O" rings are used as seals. Water, under pressure, introduced on the convex side of the diaphragm actuates the diaphragm and reverses it (turns it completely inside out upon itself about the loose circular ring).

Figure 16 shows sequence photographs of the six foot diaphragm during the first reversal. The reversal was a well controlled, rim roll mode with the diaphragm rolling through the wires, one-by-one in sequence from the first wire at the rim to the last wire at the apex. Structural performance (reversal mode and actuation pressure levels) were according to design predictions. A portion of the diaphragm actuation pressure trace (ΔP across the diaphragm) near the end of the first reversal is shown in Figure 17. Actuation pressures varied from about 1 psid start at the rim to approximately 4.5 psid at the apex at the finish of reversal.

Towards the end of reversal 1, a small leak opened up in a shell weld repair area. Testing was continued until the first diaphragm reversal was complete. The leak was repaired and reversal testing was continued subsequently as a further check of diaphragm structural performance.

The six foot diameter diaphragm was completely reversed three more times without any further leakage occurring. Diaphragm reversal modes were as well controlled as the first reversal. The condition and appearance of the diaphragm after these reversal tests was excellent. Figure 18 shows views of the diaphragm during and after the second reversal.

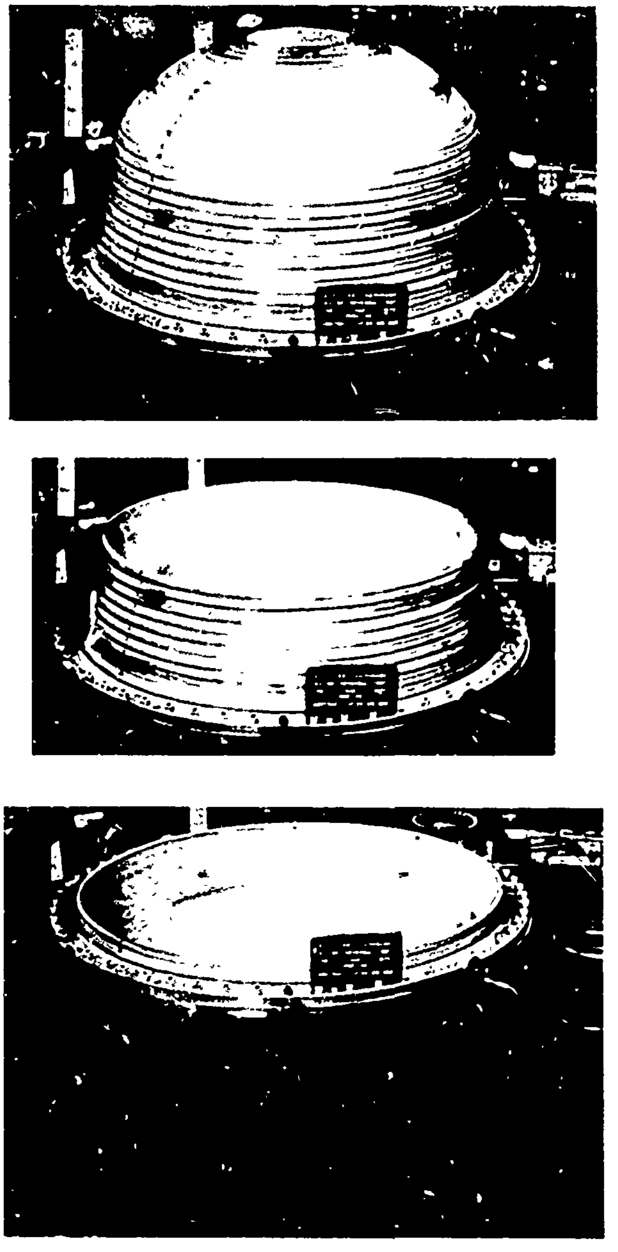


FIGURE 16. SIX FOOT DIAMETER DIAPHRAGM
FIRST REVERSAL (RIM ROLL MODE)

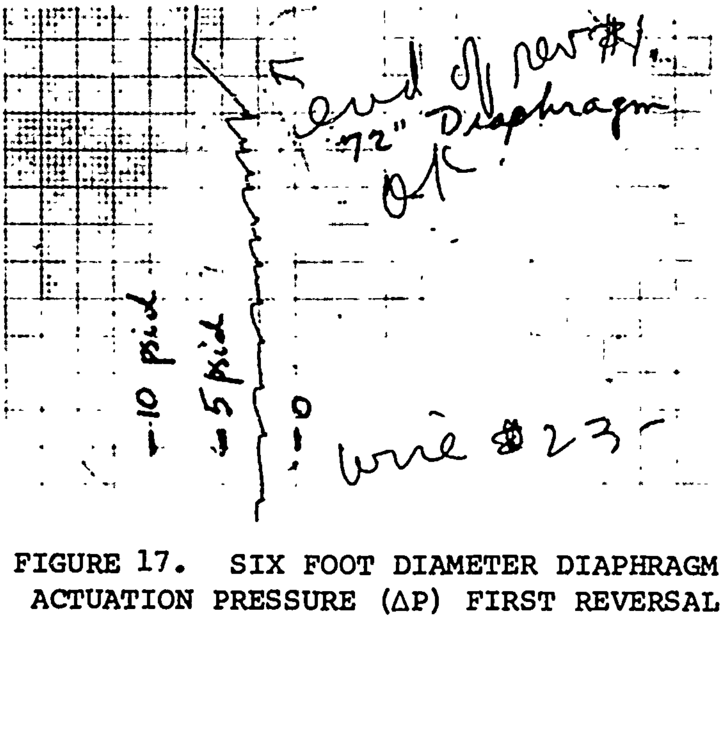


FIGURE 17. SIX FOOT DIAMETER DIAPHRAGM
ACTUATION PRESSURE (ΔP) FIRST REVERSAL

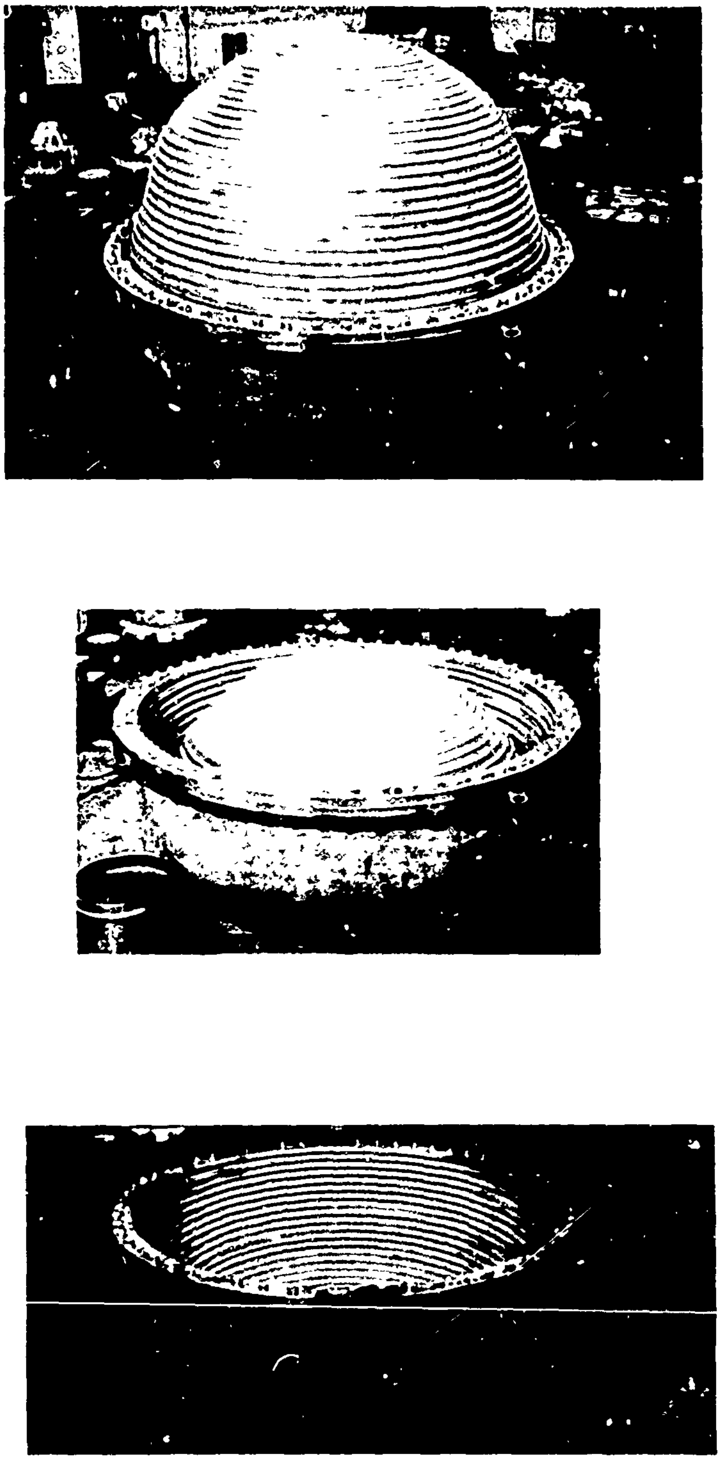


FIGURE 18. SIX FOOT DIAMETER DIAPHRAGM
SECOND REVERSAL (APEX ROLL MODE)

8. WEIGHT TRADE OFFS FOR SUBCRITICAL AND SUPERCRITICAL CRYOGENIC FLUID STORAGE AND POSITIVE EXPULSION SYSTEMS

Comparative weight studies were made for subcritical and supercritical cryogenic expulsion systems for hydrogen and oxygen. The zero g space operation mission requirements were specified by the Air Force. The results indicated that subcritical systems were lighter than supercritical systems. Aluminum tankage shows weight savings over stainless steel for subcritical application, but is heavier for the higher pressure supercritical systems.

The details of the study are given in the Summary Report contained in the Appendix (Section 11).

9. CONCLUSIONS AND RECOMMENDATIONS

9.1 Conclusions

9.1.1 Fabrication and design of stainless steel wire reinforced hemispherical type metallic diaphragms in six foot diameter size have been verified by reversal testing. Ring reinforced diaphragm scale up has now been demonstrated in the range of 1/2 to six foot diameter.

9.1.2 Use of better tooling and improved fabrication processing (particularly for tack welding the reinforcing wires to the diaphragm shell) are indicated.

9.1.3 A technique for forming large, one-piece, thin metal shells to precise contour and close thickness tolerance has been demonstrated. The fabrication method utilizes hydraulic forming coupled with plasticity theory and relatively simple tooling.

9.1.4 Use of copper plating to apply braze material to parts joined by furnace brazing has been demonstrated in six foot diameter size. This method gives precise control of the amount and distribution of braze material and simplifies braze material application.

9.1.5 Significant weight reductions are possible through the use of subcritical cryogenic fluid storage and expulsion systems compared to supercritical systems. The degree of complexity appears to be about the same for both types of systems.

Because of the low pressure and fabrication thickness limitations, aluminum tank/diaphragms are much lighter than stainless steel components for subcritical systems. Aluminum tank/diaphragms are heavier than stainless steel tankage for supercritical systems.

9.2 Recommendations

It is recommended that the demonstrated ring reinforced metallic multicycling diaphragm technology be improved and extended to meet present and contemplated needs in the areas of cost reduction, improved reliability and still larger diaphragm sizes for shuttle and other applications as outlined below.

9.2.1 Shell Forming Techniques

Start the forming process with a shell- shaped preform close to final diaphragm shell contour, which is constructed by roll and weld techniques, instead of the flat sheet starting preform used in the present program. This can result in considerable cost savings since 1) the number of forming passes and intermediate anneals are significantly reduced and 2) thin sheet rolling problems and availabiltiy in wide widths are eliminated. Moreover, for sizes larger than six feet diameter, use of welded construction is mandatory. The feasibility and reliability of welded construction for shells formed subsequent to welding has been proven out by Arde and others. The roll and welded shell preform technique utilizing the demonstrated hydraulic forming method has applications not only to diaphragms, but to tankage and other large shell-like components. Problems in availability and cost of large presses and other expensive tooling and equipment are avoided.

9.2.2 Diaphragm Assembly Methods

9.2.2.1 Development of relatively inexpensive brazing fixtures to hold the reinforcing wires in contact with the diaphragm will eliminate the need for tack welding the wires to the diaphragm. This will reduce cost and improve diaphragm reliability. A "rough cut" feasibility of this approach has been demonstrated in the present program by means of flat plate models. This effort should be continued using appropriately sized modified hemispherical type ring reinforced diaphragm brazing models.

9.2.2.2 The largest existing brazing furnaces with suitable inert atmospheres are the 100" inside working diameter vacuum furnaces utilized in the present program. For sizes larger than 100", brazing furnace availability can be a problem. Alternate methods for attachment of the reinforcing rings to the diaphragm shell should therefore be investigated. Such approaches as torch brazing (soldering) or welding would appear to be likely candidates. Arde has successfully used torch silver soldering to repair furnace brazing voids on multicycling ring reinforced diaphragms. Other torch brazing (soldering) materials may be even more suitable.

9.2.3 Alternate Materials

Effort on the present program indicates that considerable weight savings can result through the use of aluminum tank/diaphragms. The critical problem to be solved before this potential benefit can be realized is aluminum diaphragm fabrication, particularly the development of suitable methods for attachment of the reinforcing wires to the diaphragm shell. Another advantage of aluminum diaphragms is the potential for increased reversal cycle life due to aluminum's increased strain capacity to necking compared to stainless steel. Development of ring reinforced aluminum diaphragm technology, therefore, has high pay off potential.

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11. WEIGHT STUDY OF CRYOGENIC STORAGE AND EXPULSION SYSTEMS

The summary report of the weight trade off studies of subcritical and supercritical cryogenic fluid expulsion systems performed by Arde is given in this self-contained appendix. This document was previously delivered to the Air Force as required by contract.

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WEIGHT STUDY OF CRYOGENIC STORAGE AND EXPULSION SYSTEMS

1. INTRODUCTION

This preliminary summary report presents comparative estimated weights of subcritical and supercritical cryogenic expulsion systems for hydrogen and oxygen. The systems are designed for zero G space operations and meet the following requirements:

A. Hydrogen Storage System

1.	Amount stored	500 lbs.
2.	Duration of orbit service	200 hrs.
3.	Average consumption rate	2-1/2 lbs./hr.
4.	Maximum consumption rate	12-1/2 lbs./hr.
5.	Leakage or boiloff allowable	40 lb.s total max.
6.	Operating pressure	200 psia supercritical. 45 psia subcritical or slush.
7.	Storage temperature	45° R initial supercritical 45°R initial subcritical 25°R initial slush

B. Oxygen Storage System

1.	Amount stored	4000 lbs.
2.	Duration of orbit service	200 hrs.
3.	Average consumption rate	20 lbs./hr.
4.	Maximum consumption rate	100 lbs./hr.
5.	Leakage or boiloff allowable	320 lbs. total max.
6.	Operating pressure	850 psi supercritical 45 psia subcritical
7.	Storage temperature	180°R supercritical (initial) 100°R subcritical (initial)

The weight of valves, control elements, mounting structure, bosses and miscellaneous other components are not included in the calculated comparative weights since these items are beyond the scope of this study.

2. SUMMARY OF RESULTS

Table I below presents comparative weights for various subcritical and supercritical hydrogen and oxygen storage and expulsion systems.

TABLE I

		Comparative Weight (lbs.)
Group 1 - Hydrogen Stored in Stainless Steel Containers		
System 1.1	Supercritical Hydrogen	329
System 1.2	Subcritical Hydrogen	261
System 1.3	50-50 Slush Hydrogen-Helium Pressurant	224
Group 2 - Hydrogen Stored in Aluminum Containers		
System 2.1	Supercritical Hydrogen	396
System 2.2	Subcritical Hydrogen	210
System 2.3	50-50 Slush Hydrogen-Helium Pressurant	174
Group 3 - Oxygen Stored in Stainless Steel Containers		
System 3.1	Supercritical Oxygen	1080
System 3.2	Subcritical Oxygen-Helium Pressurant	239
Group 4 - Combined Oxygen and Hydrogen Stored in Stainless Steel Containers		
System 4.1	Supercritical Hydrogen & Supercritical Oxygen	1409
System 4.2	Slush Hydrogen & Subcritical Oxygen-Helium Pressurant	463

3. CONCLUSIONS

Examination of the results shows that subcritical storage systems for hydrogen and oxygen are lighter than supercritical systems for the operating requirements of this study. Slush hydrogen systems weigh slightly less than liquid hydrogen systems. The tank and bladder weights for slush hydrogen are less than for liquid hydrogen, but most of this saving is offset by the weight of the helium pressurizing system needed for slush expulsion. The great advantage of slush is the absolute avoidance of hydrogen boiloff and gas bubbles in the hydrogen tank. The disadvantage of slush systems is the complication of the added helium pressurant system.

Aluminum tankage shows substantial weight savings over stainless steel for subcritical application, but is actually heavier for supercritical systems. This is because at supercritical pressures, the tank thickness is determined by the strength/weight ratio of the material which is better for stainless steel than for aluminum. For subcritical pressures, tank thickness is determined by fabrication requirements and low density aluminum is superior. Based on fabrication requirements, the minimum shell thickness considered for this study is .015".

The subcritical systems considered are lighter overall than the supercritical. The supercritical systems require heaters and stirrers while the subcritical systems require means for pressurization. In terms of complexity, these requirements are about equal.

For oxygen, subcritical storage is very much lighter than supercritical. This is due to the high pressure required for supercritical oxygen storage which in turn calls for heavy tank walls.

Subcritical oxygen storage thus has a big built-in weight advantage over supercritical storage and can be expected to prove out lighter over a broad range of operating requirements. Subcritical storage for hydrogen also has a built-in weight advantage, but not as marked as for oxygen.

Combination storage systems for both oxygen and hydrogen may have applications for propulsion and life support in Manned Space Missions. Subcritical storage weighs substantially less than supercritical for this combination.

Aluminum construction offers additional weight advantages for subcritical storage systems and should be the subject of a more thorough study and a fabrication development program.

4. SYSTEM DESCRIPTION

System 1.1 Supercritical Hydrogen Stored in Stainless Steel Container

System 1.1 is shown schematically on Figure 1.

The system consists of a cryogenically stretch-formed stainless steel tank 73.4" diameter surrounded by a 1.6" thick layer of Linde S[illegible]-61 insulation. The tank is attached to the vehicle structure by three insulated supports. A control valve regulates the hydrogen outflow as required, a relief valve exhausts gas if the internal pressure exceeds the allowable. An electric heater maintains supercritical pressure at all times. An agitator mixes the contents to maintain uniform fluid temperature.

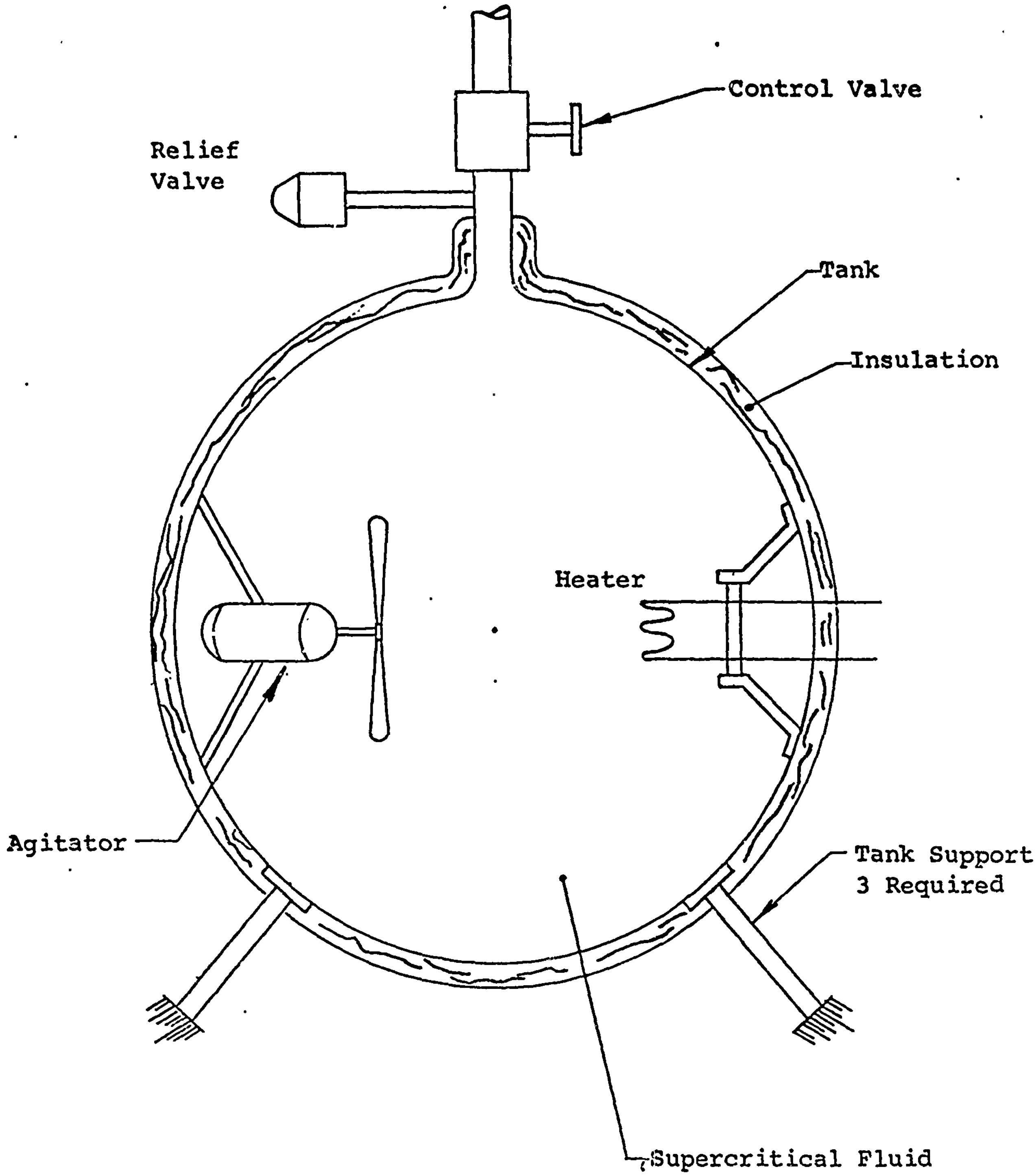


FIGURE 1. SCHEMATIC - SUPERCRITICAL HYDROGEN SYSTEM

Operation:

For hydrogen, whose critical pressure is approximately 191 psia, the minimum supercritical storage pressure is about 200 psia (Ref. 1, p.4). A typical method of flow control, and the one considered here, relies on maintaining constant pressure in the tank during operation. The required heat input per pound of hydrogen expelled in order to maintain constant pressure is shown on Figure 4 (Ref. 1, p.5) as a function of the percent of hydrogen remaining in the tank. If the heat leakage through the insulation exceeds the amount required for expulsion, the pressure will rise and gas will be lost through the relief valve. If the heat leak is less than the amount required for expulsion, the electric heater provides the necessary additional heat.

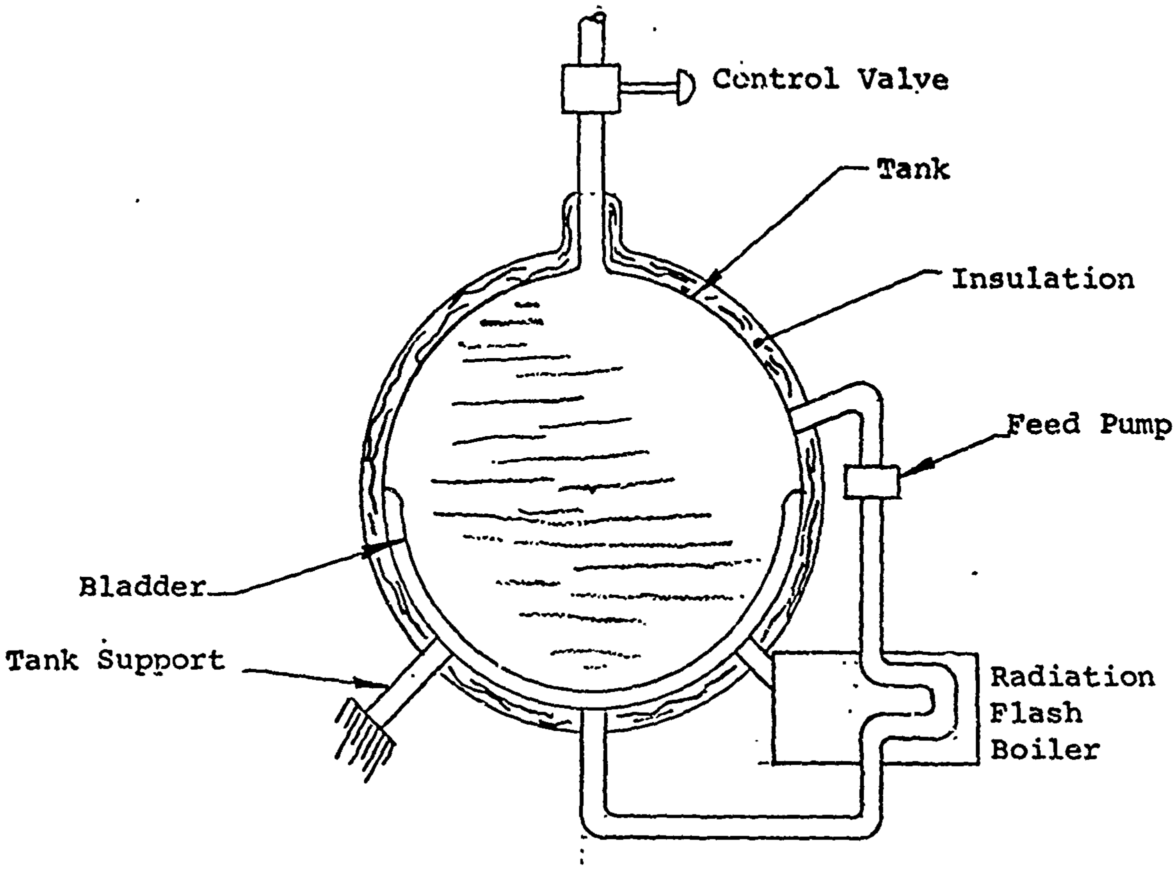
The tank is filled initially with hydrogen at 45°F and 200 psi corresponding to a density of 4.2 pounds per cubic feet. The temperature rises gradually as the tank empties.

The total comparative weight of System 1.1 is 329 pounds as follows:

Tank	173
Insulation	34
Residual Hydrogen	15
Fuel Cell & Heater	107
Total	329 pounds

System 1.2 Subcritical Hydrogen Stored in Stainless Steel Container

System 1.2 is shown schematically on Figure 2. Liquid hydrogen from the tank is pumped through a radiation flash boiler where



SCHEMATIC - SUBCRITICAL LIQUID HYDROGEN BOOTSTRAP SYSTEM

FIGURE 2

it evaporates to the pressure side of the bladder. Sufficient liquid hydrogen is pumped to maintain the tank at 45 psia. Heat leakage through the insulation will cause boiloff which is considered as part of the weight penalty. The total comparative weight of System 1.2 is 261 pounds as follows:

Tank	75	lbs.
Bladder	78	
Boiloff	17	(Ref. 4, p.5)
Pressurant	25	(Ref. 4, p.5)
Residual Hydrogen 2%	10	
Insulation	51	
Pump, Radiator	5	
Total	261	lbs.

System 1.3 50-50 Slush Hydrogen Stored in Stainless Steel Container - Helium Pressurant

System 1.3 is shown schematically in Figure 3. The system consists of a cryogenically stretch-formed stainless steel tank initially filled with 50-50 slush hydrogen. The hydrogen is expelled by a stainless steel bladder actuated by pressurized helium. The helium is retained in liquid form in an auxiliary tank equipped with a heater to maintain the helium vapor pressure at 45 psia. The helium leaving the tank passes through a solar radiation heater which heats it to 360°R. The hydrogen tank is insulated with Linde S1-62 superinsulation. The heat leaks through the insulation and from the relatively hot helium pressurant eventually converts the 50-50 slush to all liquid hydrogen, but will not cause boiloff during the operating period. The helium pressurant is heated before entering the hydrogen tank to reduce the amount required to actuate the bladder. The

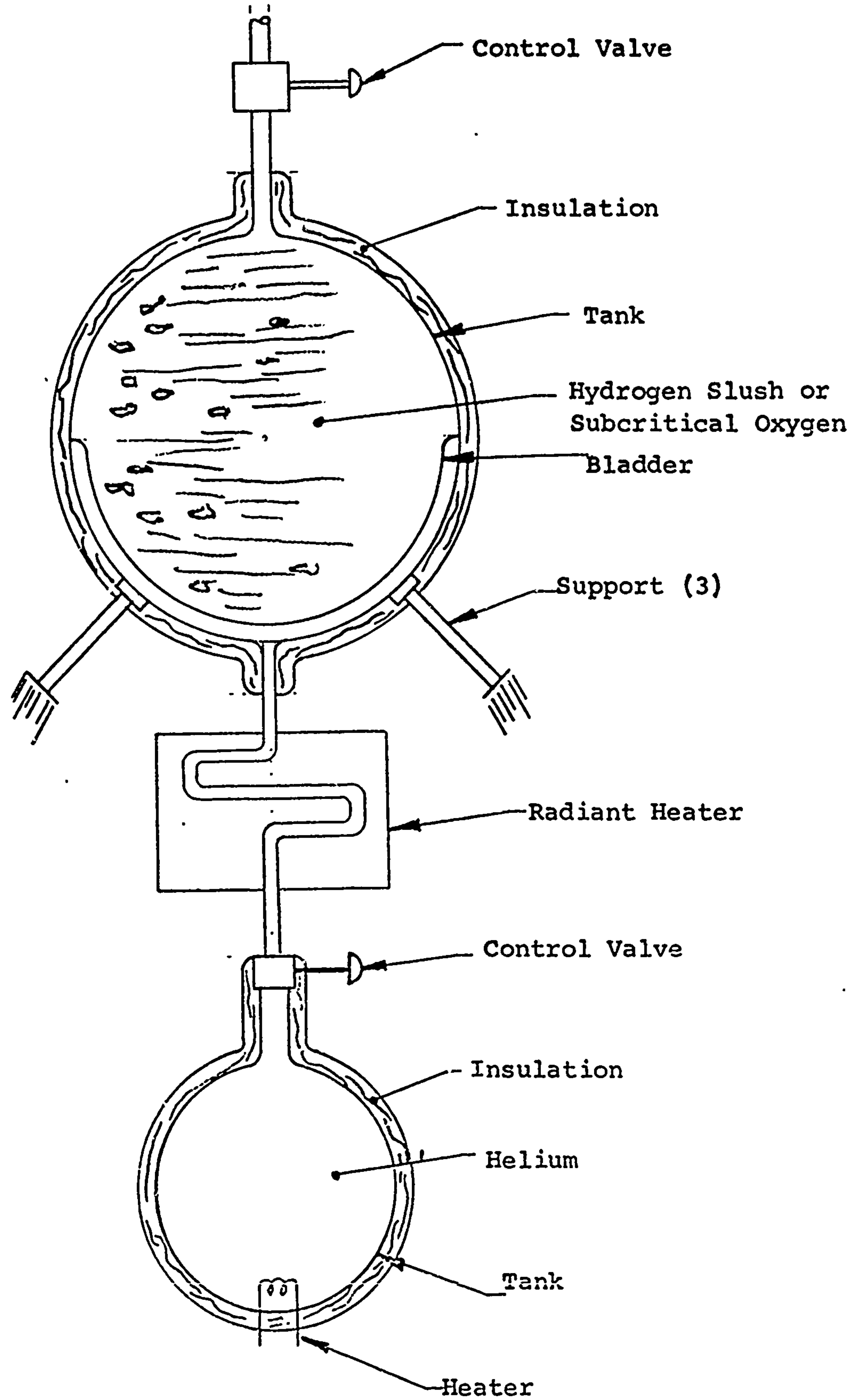


FIGURE 3. SCHEMATIC - SLUSH HYDROGEN SUBCRITICAL SYSTEM

The hotter the pressurant, the less pressurant needed. The total comparative weight of System 1.3 is 224 pounds as follows:

Hydrogen Tank	64
Bladder	65
Hydrogen Tank Insulation	24
Residual Hydrogen	10
Helium Tank	8
Helium Tank Insulation	17
Helium	33
Radiant Heater & Batteries	3
Total	224

System 2.1 Supercritical Hydrogen Stored in Aluminum Container

System 2.1 is shown schematically in Figure 1. It is the same as System 1.1 except that the tank is made of aluminum instead of high strength stainless steel. Substitution of aluminum for stainless steel simply increases the weight of the system since the strength weight ratio of aluminum is less than that of high strength stainless steel. The total comparative weight of the system is 396 pounds, as follows:

Tank	240
Insulation	34
Residual Hydrogen	15
Fuel Cell & Heater	107
Total	396

System 2.2 Subcritical Hydrogen Stored in Aluminum Container

System 2.2 is the same as System 1.2 except that the major components are of aluminum instead of stainless steel. The system

is shown schematically on Figure 2. The total comparative weight is 210 pounds as follows:

Tank	57
Bladder	45
Boiloff	17
Pressurant	25
Insulation	51
Residual Hydrogen (2%)	10
Pump, Radiator	5
Total	210

System 2.3 50-50 Slush Hydrogen Stored in Aluminum Container - Helium Pressurant

System 2.3 is shown schematically on Figure 3. It is the same as System 1.3 except that it uses aluminum components. The total comparative weight of System 2.3 is 174 pounds as follows:

Hydrogen Tank	45
Bladder	38
Hydrogen Tank Insulation	24
Residual Hydrogen (2%)	10
Helium Tank	4
Helium Tank Insulation	17
Helium	33
Radiant Heater & Batteries	3
Total	174

System 3.1 Supercritical Oxygen Stored in Stainless Steel Container

System 3.1 is shown schematically on Figure 1. The system consists of a cryogenically stretch-formed stainless steel tank 58" in diameter initially loaded with 4000 pounds of oxygen at 850 psi and 180°R temperature. The tank is insulated with Linde S1-62 insulation and is attached to the vehicle structure by three insulated supports. A control valve regulates the oxygen outflow, a relief valve exhausts gas if the internal pressure exceeds the allowable pressure. An electrical heater maintains the internal pressure at 850 psi supercritical as the oxygen is expelled. During periods of low flow demand, the heat leak through the insulation will cause the internal pressure to rise and the relief valve will blow off excess oxygen. The system comparative weight is 1080 pounds as follows:

Tank	360
Fuel Cell and Heater	675
Residual Gas	25
Insulation	20
Total	1080

System 3.2 Subcritical Oxygen Stored in Stainless Steel Container - Helium Pressurant

System 3.2 is shown schematically on Figure 3. The oxygen is loaded at 100°R and is expelled by helium gas. The system comparative weight is 239 pounds as follows:

Oxygen Tank	45
Bladder	39
Insulation (Oxygen Tank)	17
Residual Oxygen 2% (Helium Tank)	80
Insulation (Helium Tank)	17
Helium	33
Total	239

System 4.1 Supercritical Hydrogen and Oxygen Stored in Stainless Steel Containers

System 4.1 is essentially a combination of System 1.1 for hydrogen and 3.1 for oxygen. The total comparative weight is 1409 pounds as follows:

System 1.1	329
System 3.1	1080
Total	1409

System 4.2 Slush Hydrogen and Subcritical Oxygen Stored in Stainless Steel Containers - Helium Pressurant

System 4.2 is a combination of System 1.3 for slush hydrogen and System 3.2 for oxygen. The total comparative weight is 463 pounds as follows:

System 1.3	224
System 3.2	239
Total	463

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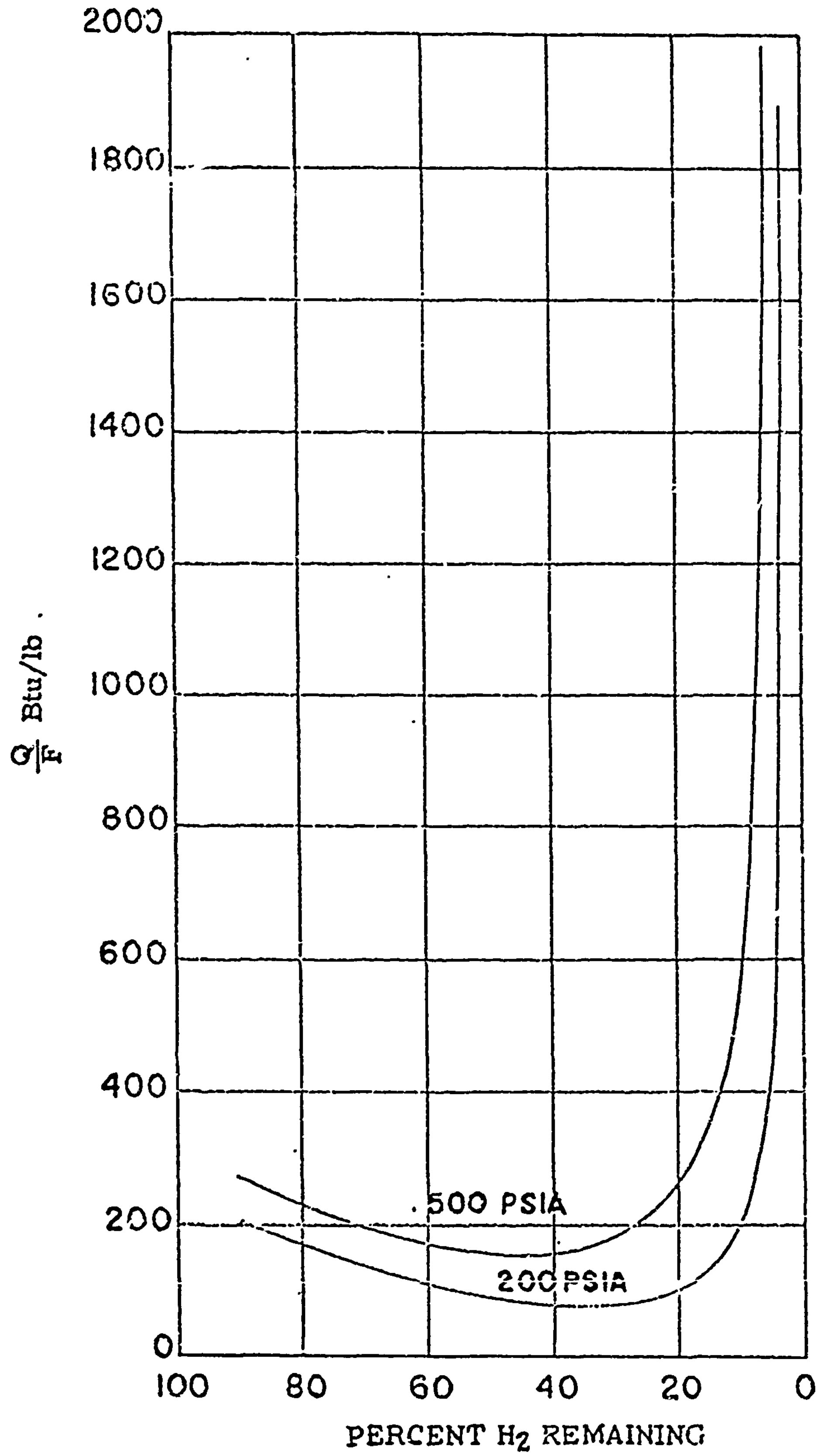

FIGURE 4 RATIO OF HEAT INPUT TO FLOW RATE VS PERCENT HYDROGEN REMAINING - SUPERCRITICAL HYDROGEN STORAGE (T_0 at 45°R)

6. CALCULATIONS

This section presents detailed calculations of the weights of the various systems described in this study. The calculations are based on the design criteria, material properties, environmental conditions and other information listed below.

1. Design Criteria for Tanks
   a. Design for burst with 1.33 safety factor.
   b. Burst pressure to be 2.2 times operating pressure.
   c. Minimum wall thickness not less than .015".
   d. Use Linde S1-62 tank insulation.
   e. Assume half of tank heat leakage comes through surface insulation, and half through supports and piping. (Ref. 5, p.57).

2. Properties of Structural Materials
   a. Cryogenically stretch-formed stainless steel - ultimate strength 300,000 psi.
   b. Aluminum 60-61T6 - ultimate strength 75,000 psi.
   c. Insulation - Linde S1-62 - Thermal Conductivity 1.8 x 10^{-5} BTU/hr.ft.°R - Density 5.5 pounds/$ft.^3$.

3. Properties of Working Fluids
   a. Supercritical Hydrogen
      At initial conditions Pressure 200 psi
Temperature 45°R
Density 4.2 lbs./$ft.^3$.
   b. Subcritical Hydrogen - Liquid
      At initial conditions Pressure = 45 psi
Temperature = 45°R
Density = 4 lbs./$ft.^3$.

c. Slush Hydrogen 50% solid, 50% liquid

Initial conditions 45 psi, 25°R

Density = 5.09 lbs./ft.^{3} (Ref. 10, p.36)

Heat required at 45 psi to melt slush and bring to boiling point 57.5 BTU/lb. (Ref. 5, p.57A)

d. Supercritical Oxygen

At initial conditions Pressure = 850 psi

Temperature = 180°R

Density = 69 lbs./ft.^{3}.

e. Subcritical Oxygen

At initial conditions Pressure = 45 psi

Temperature = 100°R

Density = 79 lbs./ft.^{3}.

f. Helium

Initial condition - Pressure = 50 psi

Temperature = 8°R

Density = 8.3 lbs./ft.^{3}.

4. Environment

Temperature 360°R.

5. Equipment Characteristics

Fuel Cell and Heater - 150 pounds overall per KW capacity.

Batteries and Heater - 10 lbs. overall per KW hour.

6. Geometric Formulae

Spherical Volume, Surface Area, Thickness.

$R = 7.44\ V^{1/3}$ inches (V in. ft.^{3})

$A = 4.83\ V^{2/3}$ ft.^{2}

NOMENCLATURE

A = surface area, inches^2 or ft.^{2}

F = pounds of hydrogen expelled

H = enthalpy, joules/gram or BTU/lb.

K = Thermal Conductivity, BTU/hr.ft.°R

KW = kilowatts

KWH = kilowatt hours

P = pressure, pounds/in^2

Q = quantity of heat, BTU

Q = heat flow per hour, BTU/hr.

q = heat flow per hour, BTU/hr.

R = radius, inches

T_i = environmental temperature, °R

T_f = fluid temperature, °R

T_g = gas temperature, °R

T = temperature, °R

t = thickness, inches or feet

V = volume, ft.^{3}

W = weight, pounds

σ = strength of material, pounds per square inch

ρ = density, pounds per cubic foot

System 1.1 Supercritical Hydrogen
Stainless Steel Container

A. Tank Weight

Tank Volume

$$V = \frac{W_h}{\rho_h} = \frac{500}{4.2} = 119 \ ft^3$$

Tank radius

$$R = 7.44 V^{1/3}$$

$$V^{1/3} = 4.92$$

$$R = 36.7 \text{ inches}$$

Surface area

$$A = 4.83 V^{2/3} = 117 ft^2 = 16{,}800 in^2$$

Wall thickness

$$t = \frac{3.72 P V^{1/3}}{\sigma} = .036 \text{ inches}$$

$$P = 200 \times 2.2 = 440 \quad psia$$

$$\sigma = 300{,}000 \div 1.33 = 225{,}000 \ psi$$

Weight

$$w = \rho A t = .286 \times 16{,}800 \times .036 = 173 \text{ lbs.}$$

E. Insulation of Tank

Previous calculations indicate that about half the heat leak into the vessel occurs through the surface insulation, the other half through supports and piping. (Ref 5 P 57)

From Fig 1 Ref 1 we can estimate that for supercritical hydrogen one pound of gas must be released per 119 BTU heat leak on the average in order to remain at 200 psi pressure. Since we are allowed 40 lbs leakage over the duration of the mission the heat leak allowable is

$$Q = 40 \text{ lbs} \times 119 \frac{BTU}{lb} = 4760 \text{ BTU}$$

over a 200 hr period. Of this one half is allocated to the surface insulation so that the allowable hourly surface heat leakage is 11.9 BTU/hr.

$$q = (T_1 - T_g)\frac{AK}{t}$$

$$t = \frac{KA(T_1 - T_g)}{q}$$

T_1 = environment temperature = 360

T_g = gas temperature = 60° average

t = insulation thickness

$$t = \frac{(360-60)1.8\times10^{-5}\times117}{11.9} = .053\ \text{ft} = .636\ \text{inches}$$

Weight of insulation

$$W = \rho A t$$
$$= 5.5\times117\times.053$$
$$= 34\ \text{lbs.}$$

c. Heating Requirement

From Ref 1 Page 5 the maximum heat input required to maintain supercritical pressure of 200 psi is 196 BTU/lb. The maximum gas outflow is 12½ lbs per hour. The maximum heat input required is 12½×196 = 2450 BTU/hr. The heat leak is 24 BTU/hr, leaving 2426 BTU/hr to be provided by

fuel cell or battery.

2426 BTU/hr = .71 KW

The fuel cell weight required for this is

$$W = .71\,KW \times 150\,\frac{\text{lbs fuelcell}}{KW} = 107\text{ lbs.}$$

Alternately, the total heat required is $119\,\frac{BTU}{lb} \times 482$ lbs expelled = 57,400 BTU

Heat Leak = $24\,\frac{BTU}{hr} \times 200 = 4800$ BTU

Net heat required = 52,600 BTU = 15.4 KWH

At 10 pounds of battery per KWH the total battery weight required is 154 pounds.

Choose fuel cell at 107 pounds

D. Residual Hydrogen

When less than 20% of the original weight of hydrogen remains, the heat input per pound of gas outflow at 200 psi begins to rise rapidly see Fig 4

When less than 10% of the hydrogen is left, the heat input from the .71 KW fuel cell is not enough to expel more hydrogen at 12½ pounds per hour at constant pressure. However, to minimize the retained hydrogen, we may abandon a policy of strict isobaric expulsion at 200 psi. An examination of Fig 5 copied from Ref 2 Fig III-1 shows that when the hydrogen density has dropped to .7 lbs per ft.^{3} i.e. 17% of the hydrogen remains, the hydrogen temperature is 70°R, well above the critical temperature of 60°R. We can then remain supercritical by expelling at constant temperature rather than constant pressure.
From Ref 1 P54 Eq I-15, the heat required to expel hydrogen at constant temperature

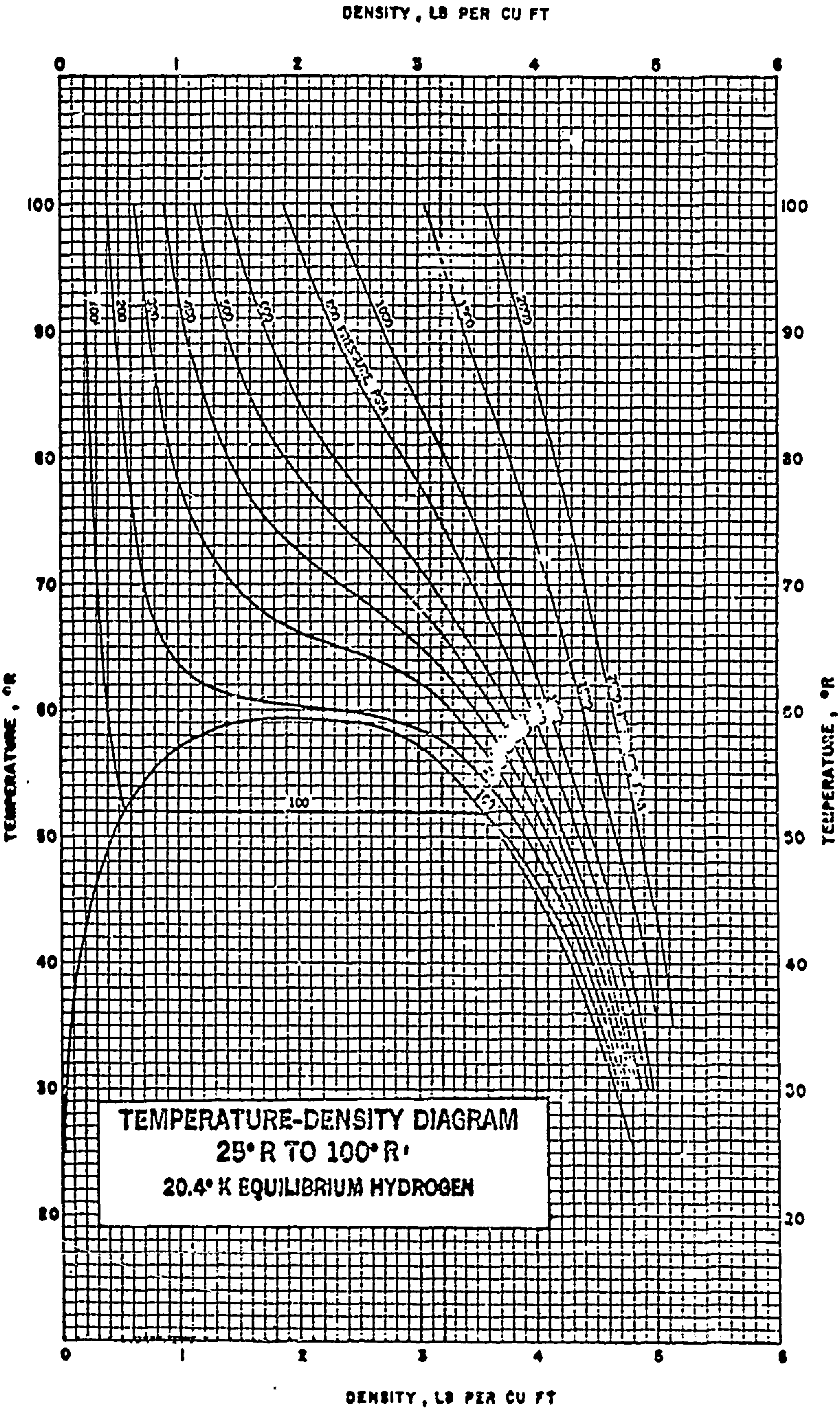
DENSITY, LB PER CU FT
TEMPERATURE, °R
100 PRESSURE PSIA
1000
2000
100
TEMPERATURE-DENSITY DIAGRAM
25° R TO 100° R
20.4° K EQUILIBRIUM HYDROGEN

FIG. 5

is

$$\frac{Q}{F} = \frac{T}{\rho}\left(\frac{\partial P}{\partial T}\right)_\rho$$

at $\rho = .7$

$$\frac{1}{\rho}\left(\frac{\partial P}{\partial T}\right)_\rho = 5.6\ \frac{ft^3\ lbs}{in^2\ lbs\ {}^\circ R} \quad \text{Ref 1 P60}$$

and $\frac{Q}{F} = 70^\circ R \times 5.6\ \frac{ft^3\ lbs}{in^2\ lbs\ {}^\circ R} \times \frac{144\ in^2}{ft^2} \times \frac{1\ BTU}{778\ ft\ lb}$

$$\frac{Q}{F} = 72.5\ \frac{BTU}{lb}$$

Expelling 12.5 lbs per hour we need

$$12.5 \times 72.5 = 905\ \frac{BTU}{hr} = .27\ KW$$

This shows we have adequate power for isothermal expulsion. Isothermal expulsion can continue until the pressure drops to 45 psia which is the minimum allowed. At 45 psia and 70°R, the residual gas density is .125 lbs per ft^3 and the weight of the residual gas is

$$W = \rho V = .125 \times 114 = 15\ lbs$$

E. Agitator

Agitation requirements for hydrogen in zero G have not been established. Consequently in this study, we will consider the agitator weight as included in the fuel cell heater expulsion system.

F. Weight Summary

Tank	173
Insulation	34
Fuel Cell, Heater Agitator	107
Residual Hydrogen	15
Total	329

System 1.1

System 1.2 Subcritical Hydrogen Bladder-Tank Bootstrap Expulsion System. Stainless Steel Tank and Bladder

A. Tank Weight

Working Pressure 45 psia

Design Temperature 45°R

Hydrogen Density 4 lbs/ft^3

Tank Volume

$V = W \div \rho_h = 500 \div 4 = 125\ ft^3$

Tank Radius

$R = 7.44\, V^{1/3} = 7.44 \times 5 = 37.2$

Tank Surface Area

$A = 4.83\, V^{2/3} = 112.1\ ft^2 = 17{,}400\ in^2$

Shell Thickness .015" minimum

Wt. of Tank

$W = \rho A t = .286 \times 17{,}400 \times .015 = 75$ lbs.

B. Weight of Bladder

The weight of a 35" radius bladder is 65.2 lbs. (Ref 3 P.6) Bladder weight varies as the cube of the radius. For a 37.2" bladder the weight will be

$$W = 65.2 \times \left(\frac{37.2}{35}\right)^3 = 78 \text{ lbs}$$

C. Boil Off 17 lbs (Ref 4 P5)

D. Pressurant 25 lbs (Ref 4 P5)

E. Unexpelled LH_2 10 lbs

This is based on 98% expulsion efficiency of bladder

F. Insulation

Use 1" Linde SI-62

$$\text{Volume Insulation} = A \cdot t = 112.1 \times \frac{1}{12} = 9.34 \text{ ft}^3$$

$$W = \rho V = 5.5 \times 9.34 = 51 \text{ lbs}$$

G. Radiation Heater - 3 pounds maximum

H. Bootstrap Pump - 2 pounds estimated

System Weight - 261 pounds - Total

System 1.3 — 50-50 Slush Hydrogen
Helium Pressurized
Stainless Steel Vessels

A. Tank- (Hydrogen)

$$\text{Volume} = \frac{W}{\rho} = \frac{500}{5.09} = 98.5\ ft^3$$

Tank Radius = $7.44\ V^{1/3} = 34.5''$

Surface Area = $694\ V^{2/3}\ in^2 = 14850\ in^2 = 103\ ft^2$

Wall thickness = .015"

Wt of Tank = $At\rho = 14850 \times .015 \times .286 = 64$ lbs

B. Tank Insulation

Try 1/2" thick Linde SI-62 and see how much boiloff if any occurs.

Heat leak through insulation

$$Q = \frac{K}{t} A (T_i - T_f)$$

$$Q = \frac{1.8 \times 10^{-5}}{1/24} \times 103 (360 - 25) = 15\ BTU/hr$$

Heat leak through support + piping

15 BTU/hr

Over 200-hour service leak through insulation, supports and piping = 6000 BTU.

Heat Leak from Helium Pressurant
Helium pressurant enters at 360°R
Weight of Pressurant - 25 lbs Ref 6 P356
Final average pressurant
temperature = 90° [Ref 6 P356, Ref 11 P1032]
Heat transferred from pressurant to slush

$$Q = w \; C_p \, \Delta T$$
$$= 25 \times 1.25 (360 - 90)$$
$$= 8400 \text{ BTU}$$

Total Heat Input from insulation, support, piping and hot pressurant 14,400 BTU

Heat required to melt slush, and bring to boiling point
57.5 BTU/lb.
Average amount of hydrogen in tank during mission - 250 lbs

Total heat required to bring average amount of slush to boiling point

$$Q = \bar{W} \times 57.5 = 250 \times 57.5 = 14{,}400 \text{ BTU}$$

Conclusion: With ½" insulation there will be substantially no boil off due to heat leakage.

Wt. of ½" insulation

$$W = \rho A t = 5.5 \times 103 \times \frac{1}{24} = 24 \text{ lbs.}$$

C. Wt of Bladder

65 lbs. Ref 7 P 39

D. Helium Tank

Pressure - 50 psi

Temp. 8°R

Density 8.3 lbs/ft³ Ref 9 P 28

Enthalpy 5.04 BTU/lb "

Try a 4 cubic foot tank

$R = 7.44\, V^{1/3}$ inches $= 7.44 \times 1.59 = 11.85$"

Area $= 4.83\, V^{2/3}$ ft² $= 4.83 \times 2.52 = 12.2$ ft² $= 1760$ in²

Wt of tank

$$W = \rho A t = .286 \times 1760 \times .015 = 7.5 \text{ lbs}$$

Wt of Helium $= 4 \times 8.3 = 33$ lbs

E. Helium Tank Insulation

Try 3" thickness Linde SI-62

Wt of Insulation

$$W = \rho A t = 5.5 \times 12.2 \times \frac{1}{4} = 17 \text{ pounds.}$$

Heat Leak

$$Q = A \Delta T \frac{K}{t}$$

$$= 12.2(360-8)\frac{1.8 \times 10^{-5}}{1/4} = .32 \text{ BTU/hr}$$

Leak through supports + pipes .32 BTU/hr

Total heat leakage = .64 BTU/hr

F. Boil off

Worst condition if no hydrogen expulsion for 160 hours. All expulsion of hydrogen during last 40 hours at 12½ lbs/hr.

Heat leak over 160 hours

$$Q = 160 \times .64 = 102 \text{ BTU}$$

Heat leak per pound of helium

$$Q/w = 102/33 = 3 \text{ BTU/lb}$$

Enthalpy of hot helium = 5.04 + 3 = 8.04 BTU/lb.

Since the volume remains constant we determine from Ref 9 P 40 + 42 that the pressure of the helium

is now over 120 and less than 140 psi.
The allowable tank pressure is

$$P = \frac{2\sigma t}{R}$$

applying the operating to burst stress ratio of 2.2 and a safety factor of 1.33 we have

$$P = \frac{2 \times 300{,}000 \times .015}{1.33 \times 2.2 \times 11.85} \cdot 260 \text{ psi}$$

We can therefore retain all the helium by permitting the pressure to build up

G. Helium Expulsion
The average amount of heat required to expel helium at constant pressure is
9 BTU per pound Ref 7 P 164
The total required for 25 pounds is $25 \times 9 = 225$ BTU.
The heat leak is $200 \times .64 = 128$ lbs.
The difference $225 - 128 = 97$ BTU must be supplied by batteries

97 BTU = (97 ÷ 3413) KWH = .03 KWH
At 10 pounds per KWH, the battery weight = .3 lbs say 1 lb.

H. Radiant Heater to heat pressurant entering hydrogen tank 2 pounds.

I. Weight Summary

Hydrogen Tank	64
Bladder	65
Hydrogen Tank Insulation	24
Residual Hydrogen (2%)	10
Helium Tank	8
Helium Tank Insulation	17
Helium	33
Radiant Heater	2
Batteries and Internal, Heater	1
System 1.3 Total	224 lbs

Note: The helium tank and its insulation are oversize by about 20% since we need 25 lbs of helium and we are providing 33 lbs.

Case 2.1 Supercritical Hydrogen Stored in Aluminum Container

A. Tank

Radius $R = 36.7$ Ref. Case 1.1

Surface Area $16{,}800 in^2$ Ibid

Design pressure $= 200 \times 2.2 = 440$ psi

Design strength

$$\sigma = 75{,}000 \div 1.33 = 56{,}250$$

Wall Thickness

$$t = \frac{PR}{2\sigma} = \frac{440 \times 36.7}{2 \times 56{,}250} = .143$$

Weight of tank

$$W = \rho A t = .1 \times 16{,}800 \times .143 = 240 \text{ lbs}$$

B. Weight Summary

Tank	240	
Insulation	34	Ibid
Residual Hydrogen	15	"
Fuel Cell, Heater and Agitator	107	"
System 2.1 Total	396	lbs

System 2.2 Subcritical Hydrogen
Aluminum Container

A. Tank Weight

Radius $R = 37.2$ See System 1.2

Wall Thickness

$$t = \frac{PR}{2\sigma}$$

$$P = 45 \times 2.2 = 99$$

$$\sigma = 75{,}000 \div 1.33 = 56{,}250$$

$$t = \frac{99 \times 37.2}{2 \times 56250} = .0328$$

Surface Area

$A = 17{,}400 \text{ in}^2$ Ibid

Weight of Tank

$W = \rho A t = .1 \times 17{,}400 \times .0328 = 57$ lbs.

B Bladder

Per Ref 7 P 169 the weight of an aluminum bladder is

$$W_a = \frac{W_s}{1.73}$$

where W_s is the weight of a stainless steel bladder. From TB System 1.2 $W_s = 78$ lbs. $W_a = \frac{78}{1.73} = 45$ lbs.

c. Weight Summary

Tank	57	
Bladder	45	
Boiloff	17	See System 1.2 Preceding
Pressurant	25	
Unexpelled LH_2	10	
Insulation	51	
Radiation Heater	3	
Pump	2	
System 2.2 Total Weight	210	

System 2.3 50-50 Slush Hydrogen
Helium Pressurant
Aluminum Container

A Hydrogen Tank

Tank Radius $R = 34.5''$

Operating Pressure 45 psi

Design pressure $45 \times 2.2 = 99$

Working strength of aluminum 56,250 (See System 2.1 II A)

Wall thickness

$$t = \frac{PR}{2\sigma} = \frac{99 \times 34.5}{2 \times 56250} = .030$$

Surface area = 14850 in^2 (Ref System 1.3)

Weight of tank

$$W = \rho A t = .1 \times 14850 \times .030 = 44.6 \text{ lbs}$$

B Weight of Bladder

Weight of steel bladder 65 lbs. (Ibid)

Weight of equivalent aluminum bladder

$$W_a = \frac{W_s}{1.73} \quad \text{Ref 7 P 169}$$

$$W_a = \frac{65}{1.73} = 38 \text{ lbs}$$

C. Helium Tank

$R = 11.85''$ (Ibid)

$P = 50$ (Ibid)

Take $t = .020''$ sufficiently strong

Surface area = 1760 in^2 Ibid

Wt of tank

$$W = \rho A t = .1 \times 1760 \times .020 = 3.5 \text{ lbs}$$

D. Weight Summary

Hydrogen Tank	45	
Bladder	38	
Hydrogen Tank Insulation	24	System 1.3
Residual Hydrogen	10	"
Helium Tank	4	
H Tank Insulation	17	System 1,3
Helium	33	
Radiant Heater	2	
Batteries and Internal Heater	21	
System 2,3 Total Weight	174 lbs	

System 3.1 Supercritical Oxygen
Stainless Steel Container

A. Tank Weight

Operating Pressure	850 PSI	
Initial Temperature	180°R	
Gas specific gravity	1.1	Ref 2 Fig VII-1
Gas density	69 lbs/ft^3	
Wt of gas carried	4000 lbs	

Volume $V = \frac{4000}{69} = 58\ ft^3$

Radius $R = 7.44 V^{1/3}$ inches $= 28.9$ inches

Surface Area $= 695 V^{2/3}\ in^2 = 10{,}450\ in^2$

Design pressure $P = 2.2 \times 850 = 1870$ psi

Material Strength

$$\sigma = 300000 \div 1.33 = 225{,}000$$

Wall thickness

$$t = \frac{PR}{2\sigma} = \frac{1870 \times 28.9}{2 \times 225{,}000} = .121$$

Weight

$$W = \rho A t = .286 \times .121 \times 10450 = 360\ lbs$$

B. Heating Requirement

Estimate heat required to expel oxygen at constant pressure. For convenience consider an initial pressure of 868 psi. rather than 850 so that for estimating purposes we can use the 60 atmosphere table of Ref 12 directly without interpolation.

We have to start

$T = 180°R = 100°K$

p = 60 atmospheres

ρ = 1.104 gms/cc Ref 12 P. 70

$H \cdot$ 160.91 joules/gm

Suppose we begin with 1.104 gms oxygen in 1 cc volume. We now expel .0155 gms of oxygen at constant pressure leaving 1.0885 gms at 60 atmosphere pressure. The enthalpy of the residue is 166.12 joules/gm.

The change of enthalpy is 166.12 − 160.91 = 5.21 joules/gm. The average amount of gas in the 1cc volume during this process was 1.0962 gms

The heat added was

$$Q = \bar{W}(\Delta H) = 1.0962 \times 5.21$$

And the heat added per gram of gas expelled was

$$q = \frac{\bar{W}\,\Delta H}{\Delta W} = \frac{1.0962 \times 5.21}{.0155} = 367 \frac{\text{joules}}{\text{gm expelled}}$$

Now

$$1\frac{\text{joule}}{\text{gm}} = \frac{\text{joules}}{\text{gm}} \times \frac{9.48 \times 10^{-4}\ \text{BTU}}{\text{joule} \times \text{lb}} \times 453\ \text{gms} = .429 \frac{\text{BTU/lb}}{\text{joules/gm}}$$

$$q = 367 \frac{\text{joules}}{\text{gm expelled}} = 154 \frac{\text{BTU}}{\text{lb expelled}}$$

The maximum required rate of expulsion is 100 lbs/hour so we must be able to supply 15,400 BTU/hr to expel initially at constant pressure This is equal to 4.5 KW capacity At 150 pounds per KW, the heater weight is

$$W = 4.5 \times 150 = 675 \text{ pounds.}$$

The average heat required for expulsion from full to near empty is 61.5 BTU/lb. (Ref 7 P170) For 4000 lbs this is 246,000 BTU or 72 KWH

at 10 lbs per KWH for batteries, a battery supply would weigh 720 pounds. Since this is more than the fuel cell weight we will select the fuel cell at 675 pounds.

C. Residual Gas

The minimum pressure end condition is 45 psi. Assuming we stay above the critical temperature at say 300°, the final gas specific gravity is .0071 and the density is .443 lbs/ft^3

The weight of the residual gas is

$$W = \rho V = .443 \times 58 = 25 \text{ lbs}. \text{ (Ref 2 Fig VII-1)}$$

D Insulation - Use .6" Linde S1-62

$$W = \rho A t = 5.5 \times \frac{10450}{144} \times \frac{.6}{12} = 20 \text{ lbs.}$$

E. Weight Summary

Tank	360
Insulation	20
Residual Oxygen	25
Agitator, Fuel Cell & Heater	675
System 3.1 Total Weight	1080 lbs

System 3.2 Subcritical Oxygen
Stainless Steel Container
4000 lbs. oxygen

A. Tank Weight

Operating Pressure 45 psi

Initial Temperature 100°R

Density - use 69 lbs/ft^3.

Volume = 58 ft^3

Radius = 28.9"

Surface area = 10,450

Wall thickness = .015 minimum

Weight

$W = \rho A t = .286 \times .015 \times 10450 = 45$ lbs

B Bladder Weight

Bladder weight varies as volume

For V = 97.5, bladder weight = 65 Ref 7 P139

For V = 58 ft^3

$W = \left(\frac{58}{97.5}\right) \times 65 = 39$ lbs

c Insulation

Use ½" Linde SI-62

Heat leak

$$Q = \frac{K}{t} A(T_i - T_F)$$

$t = \frac{1}{24}$ ft

$K = 1.8 \times 10^{-5}$

$A = 10450 \text{ in}^2 = 72.5 \text{ ft}^2$

$T_i = 360°R, \quad T_F = 100°R$

$$Q = \frac{1.8 \times 10^{-5}}{1/24} \times 72.5(360-100) = 8.1 \text{ BTU/hr}$$

Assume equal heat leak
through supports + fittings = 8.1 BTU/hr
Total heat leak 16.2 BTU/hr
Over 200 hours 3220 BTU.

Effect of heat leak
Enthalpy of Oxygen at 100°R = 93.3 joules/gm
Enthalpy of oxygen at boiling point a 45 psi 160 joules/gm
Change in enthalpy = 66.7 joules/gm
= 28.5 BTU/lb

Heat req to bring 4000 lbs oxygen
to boiling point = 4000 × 28.5 = 114000 BTU
Heat leakage negligible

Wt. of insulation

$$W = \rho A t = 5.5 \times 72.5 \times \frac{1}{24} = 16.5 \text{ lbs.}$$

D. Other Components

We now conservatively assume the weight of the helium pressurizing elements to be the same as for System 1.3.

E. Weight Summary

Oxygen tank	45
Oxygen bladder	39
Oxygen tank insulation	17
Residual oxygen 2%	80
Helium Tank	8
Helium Tank Insulation	17
Helium	33
System 3.2 Total weight	239 lbs.

System 4.1 Supercritical Hydrogen and Oxygen in Stainless Steel Containers.
500 lbs hydrogen
4000 lbs oxygen

System 4.1 is essentially a combination of Systems 1.1 and 3.1 for hydrogen and oxygen respectively

A. Weight Summary

System 1.1	329
System 3.1	1080
Total System 4.1	1409

<u>Systems 4.2</u> Subcritical Hydrogen and Subcritical Oxygen Stored in Stainless Steel Containers
500 lbs hydrogen
4000 lbs oxygen

System 4.2 is a combination of Systems 1.3 and 3.2 for slush hydrogen and oxygen respectively.

A. Weight Summary

System 1.3	224
System 3.2	239
Total System 4.2	463 lbs.

END

DATE

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